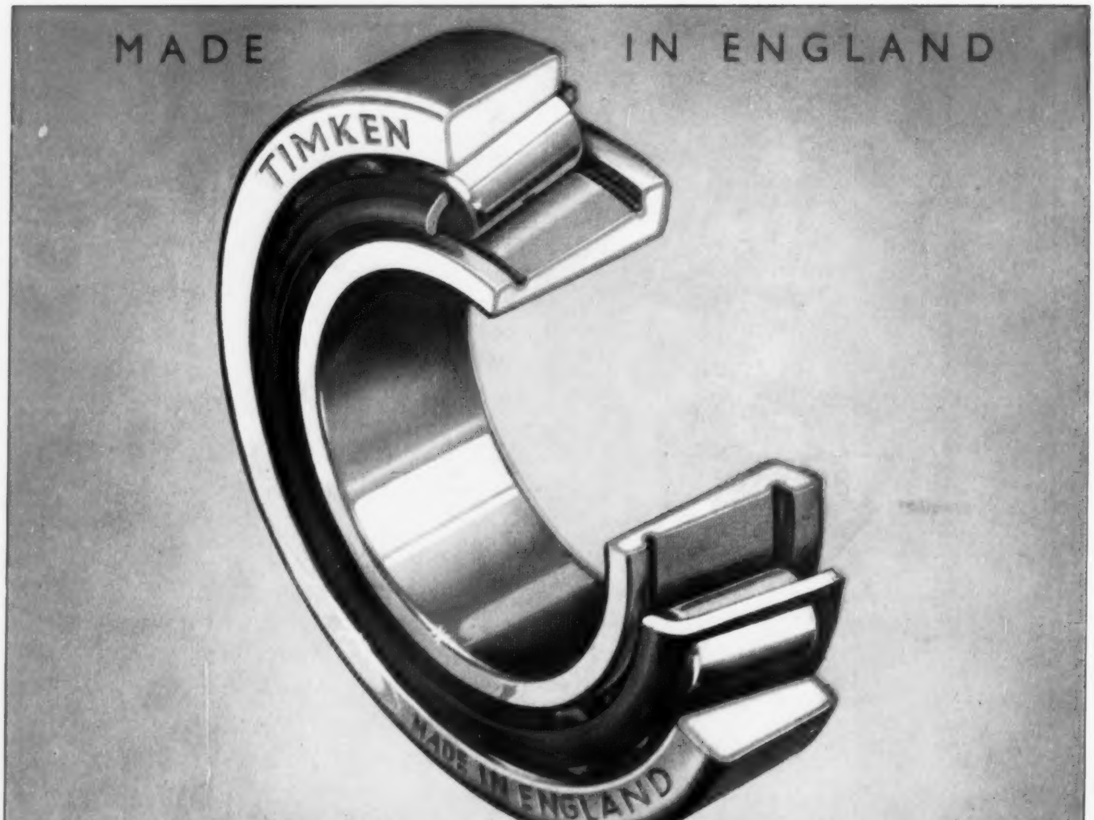
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 44 No. 3 MARCH 1954 PRICE: 3s. 6d.



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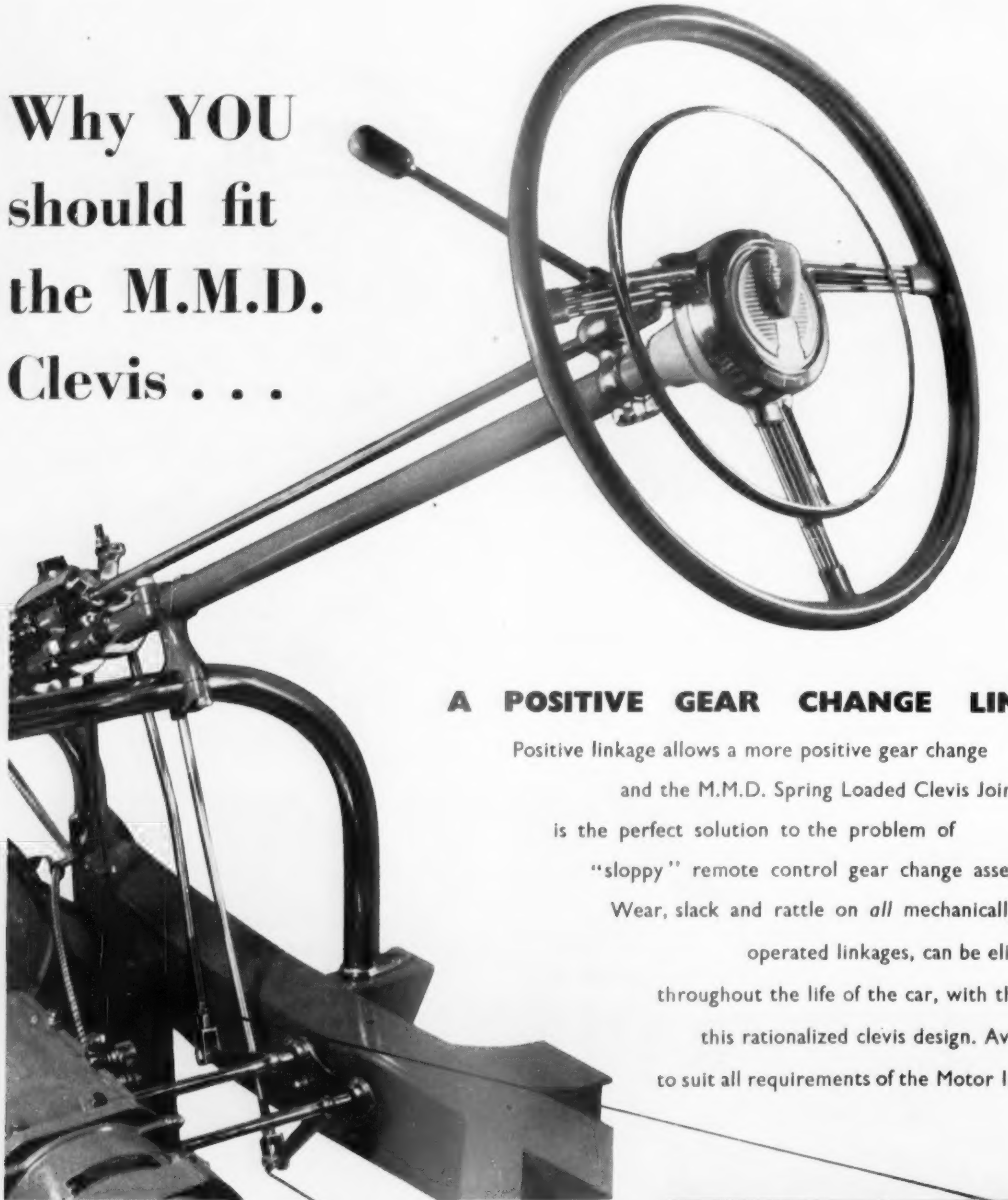

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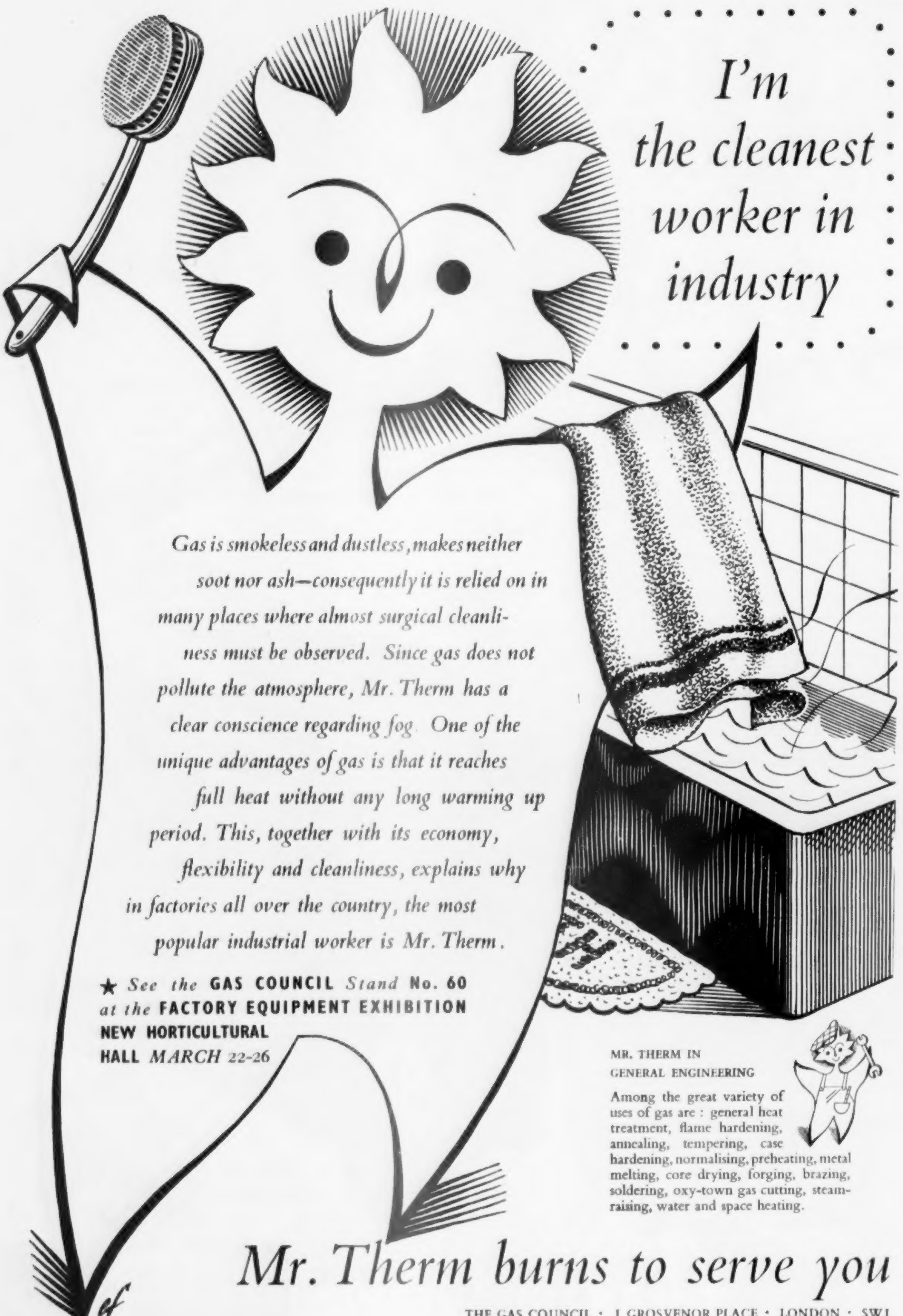
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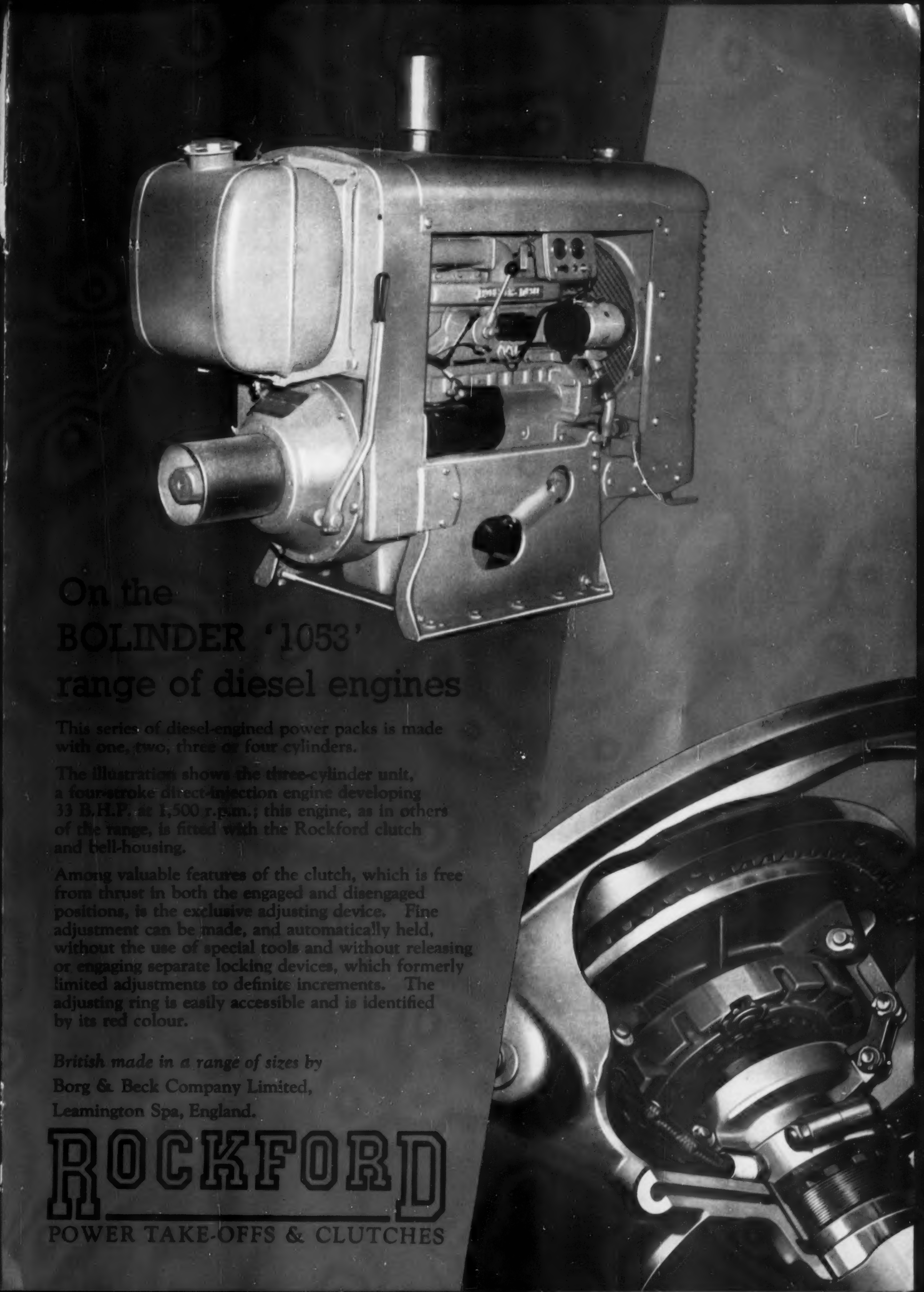
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This series of diesel-engined power packs is made with one, two, three or four cylinders.
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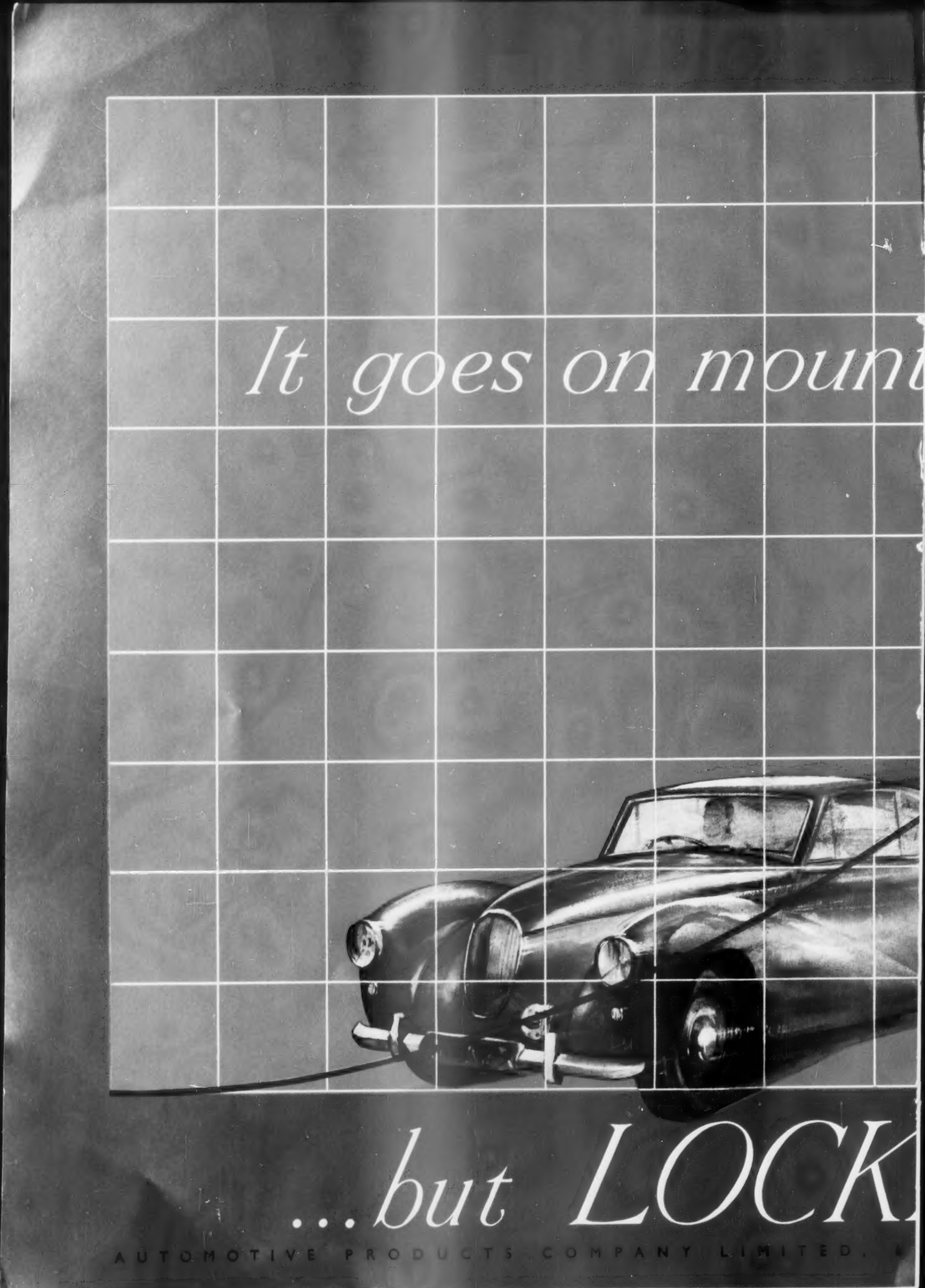

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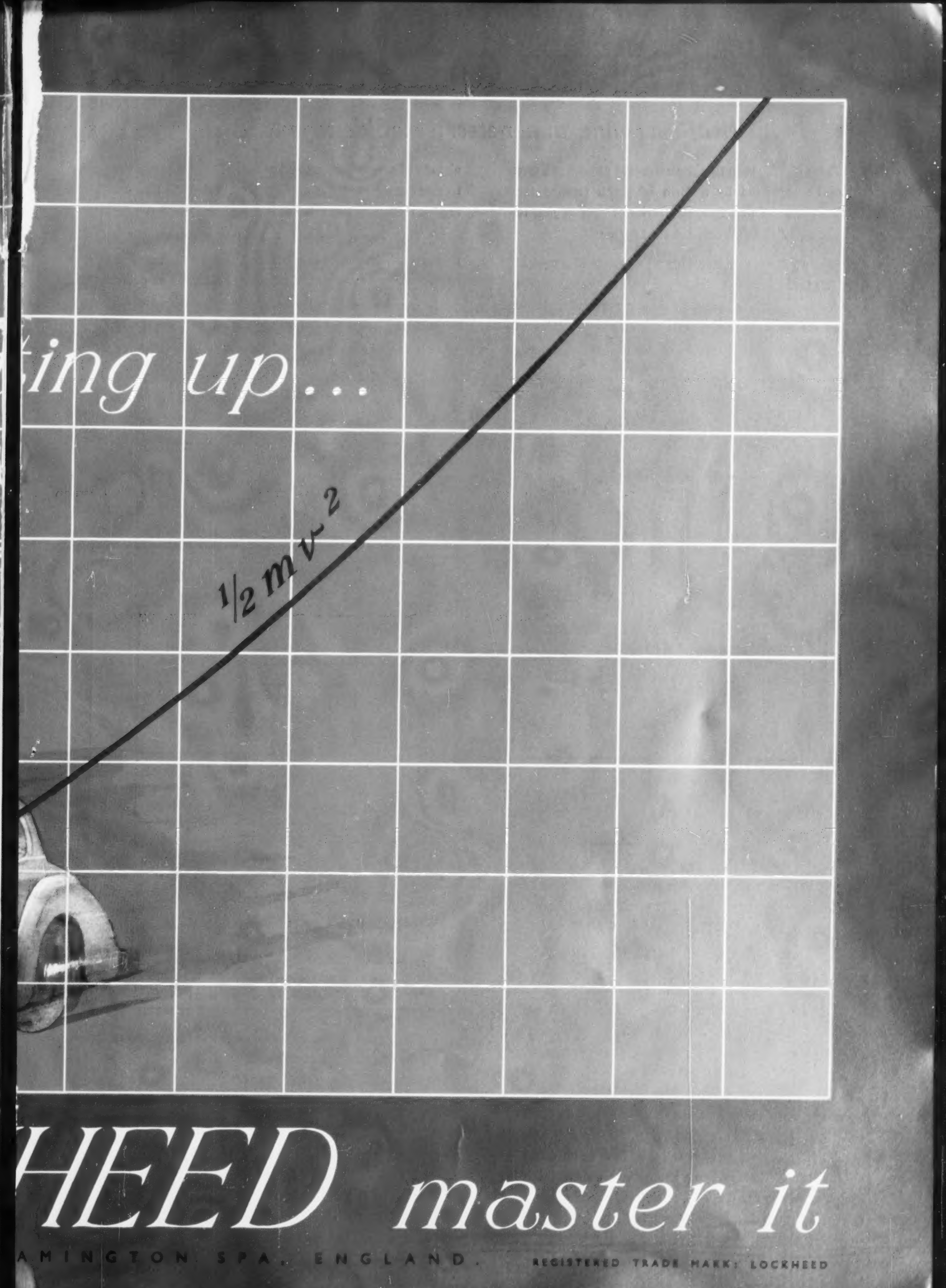

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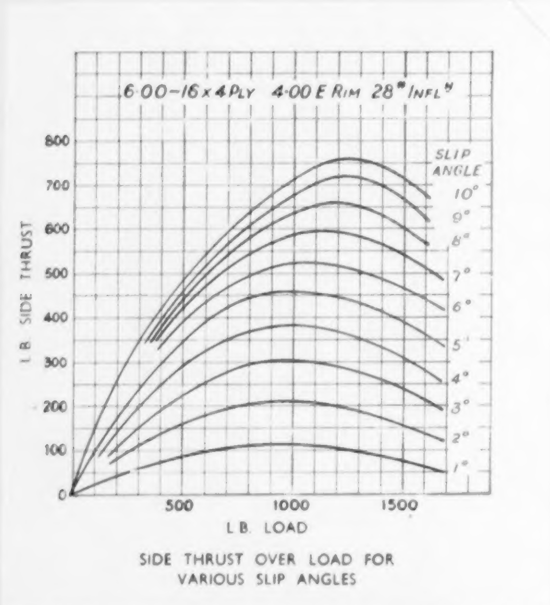
SUBTLETIES OF STEERING

Causes of over- and under-steer: 8. Sideways load (b)

WE have already seen that the cornering powers of a tyre can be expressed by a form of graph relating the cornering power of that tyre, at a series of drift angles, to the load carried (Olley, Proc. I.A.E., 1947).

Suppose we take such a graph and mark on it a vertical line at the mean load per tyre of the pair of wheels we are considering. If there were no weight transference we could just read off the drift angles for different cornering forces



The basic diagram showing how side thrust in cornering power varies with the load carried by a tyre for different drift angles.

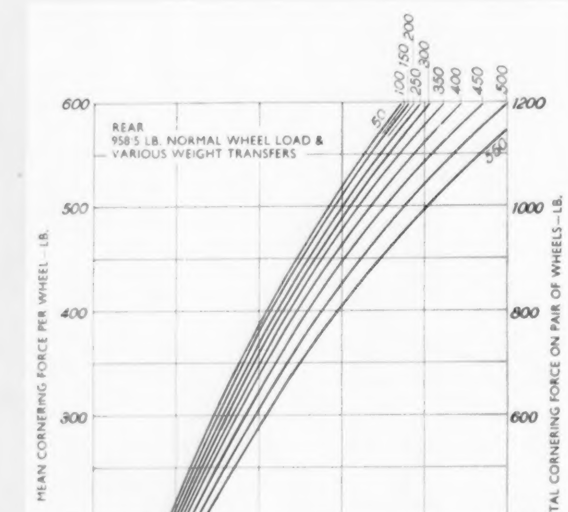
and plot them as a further graph of drift angle against cornering force, for zero weight transference.

This is all very fine, but we know that weight transference does occur. Let us therefore next draw in two further lines, on our original graph, one at a load 50 lb. more than our mean, the other at a corresponding load 50 lb. less. The sum of the cornering forces at which these two lines cut the 2° drift angle line give us the total cornering force per pair of wheels at the weight transference of 60 lb. and the drift angle of 2°; further total cornering forces for other drift angles can be obtained and plotted as a further line on our second graph, labelled for 50 lb. weight transference. Other similar lines for 100 lb., 150 lb. and so on of weight transference are similarly arrived at and can be plotted on the same graph (which, by its mean load, will apply to the front or rear of the car we are considering; two such graphs are required for considering one car). If by now we have worked out the total sideways loads and weight transferences for each end of the car for various sideways accelerations, we can arrive at the drift angles, front and rear, at the different sideways accelerations or severities of cornering. We will already of course have arrived at the roll angles appertaining to these same sideways accelerations. What do these drift angles, front and rear, mean?

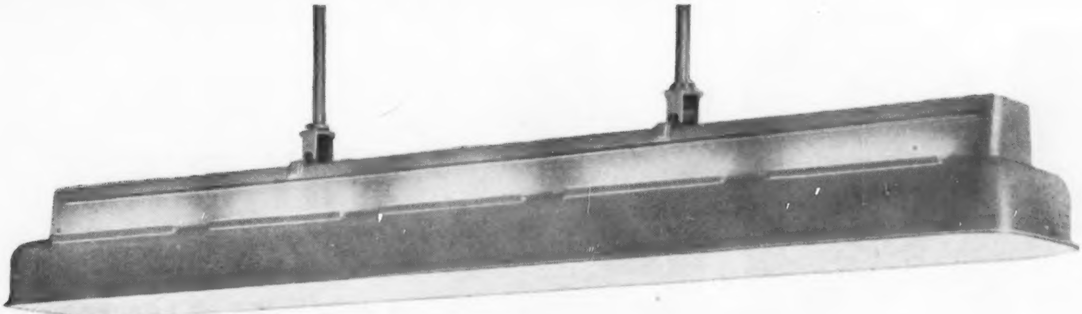
First, of course, we must modify them by any amount of roll steer which exists; this has already been discussed in an earlier essay, and where it exists will increase or decrease the drift angles concerned.

To take a car of average wheelbase round a 108 ft. radius will require a steering angle of the front wheels of about 5°.

If the front and rear wheels have equal effective drift angles, then the vehicle will still be turning in a 108 ft. radius but the centre of that radius will have moved by an amount 108 times the drift angle in radians. If, however, the front drift angle, for example, is greater than the rear, then the front wheels will have to be turned by an extra amount, equal to the difference between the two drift angles, in order



to maintain the radius of turn, and the car is said to under-steer by the amount of the difference in the effective drift angles. Similarly, a greater rear drift angle means an over-steer of the amount by which this rear drift angle exceeds the front.



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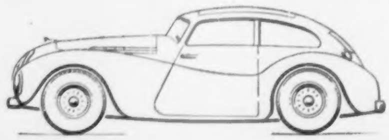

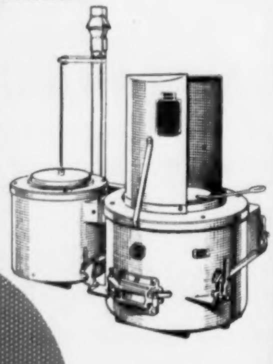

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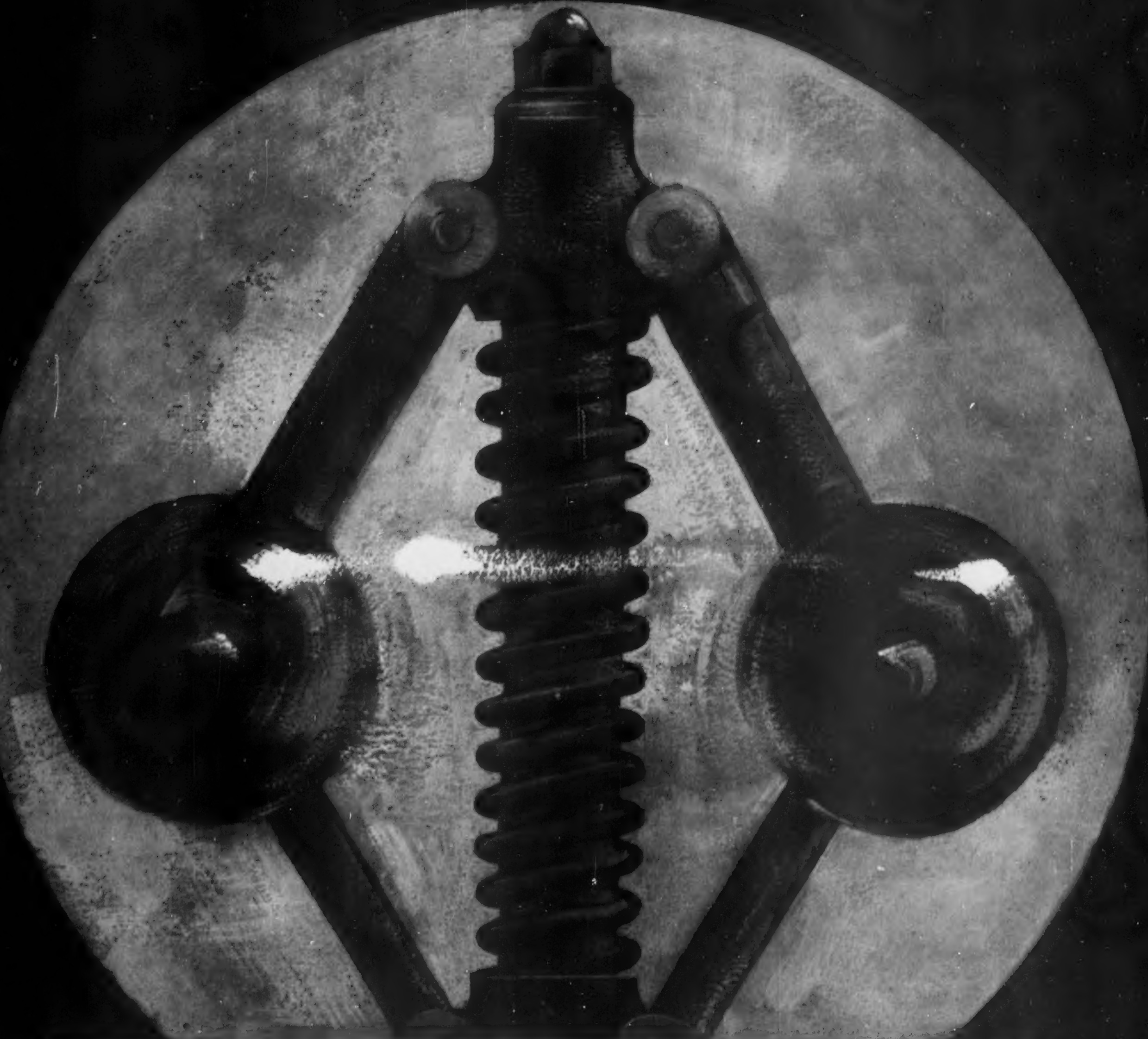

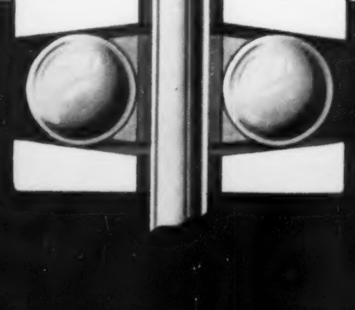

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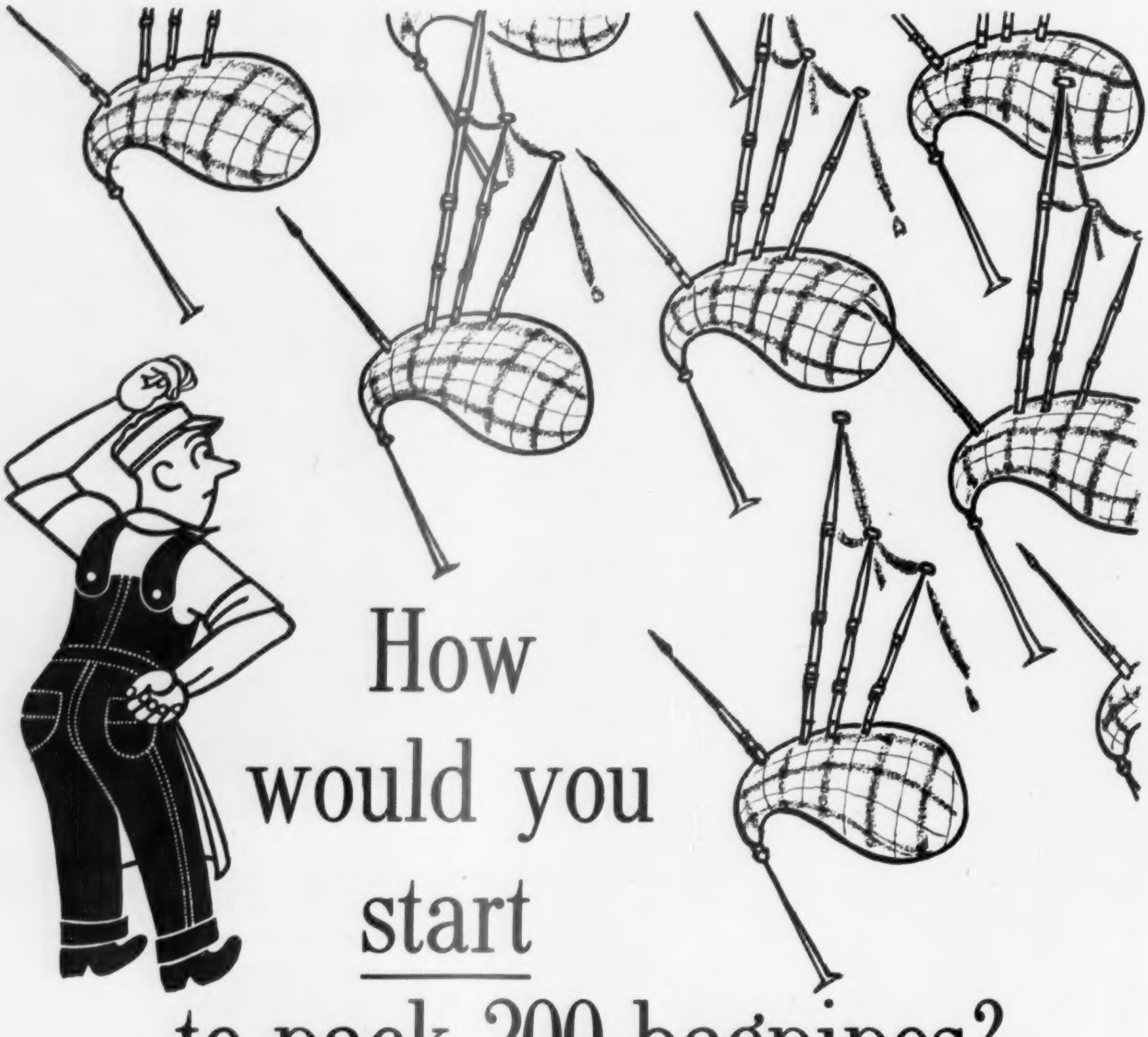

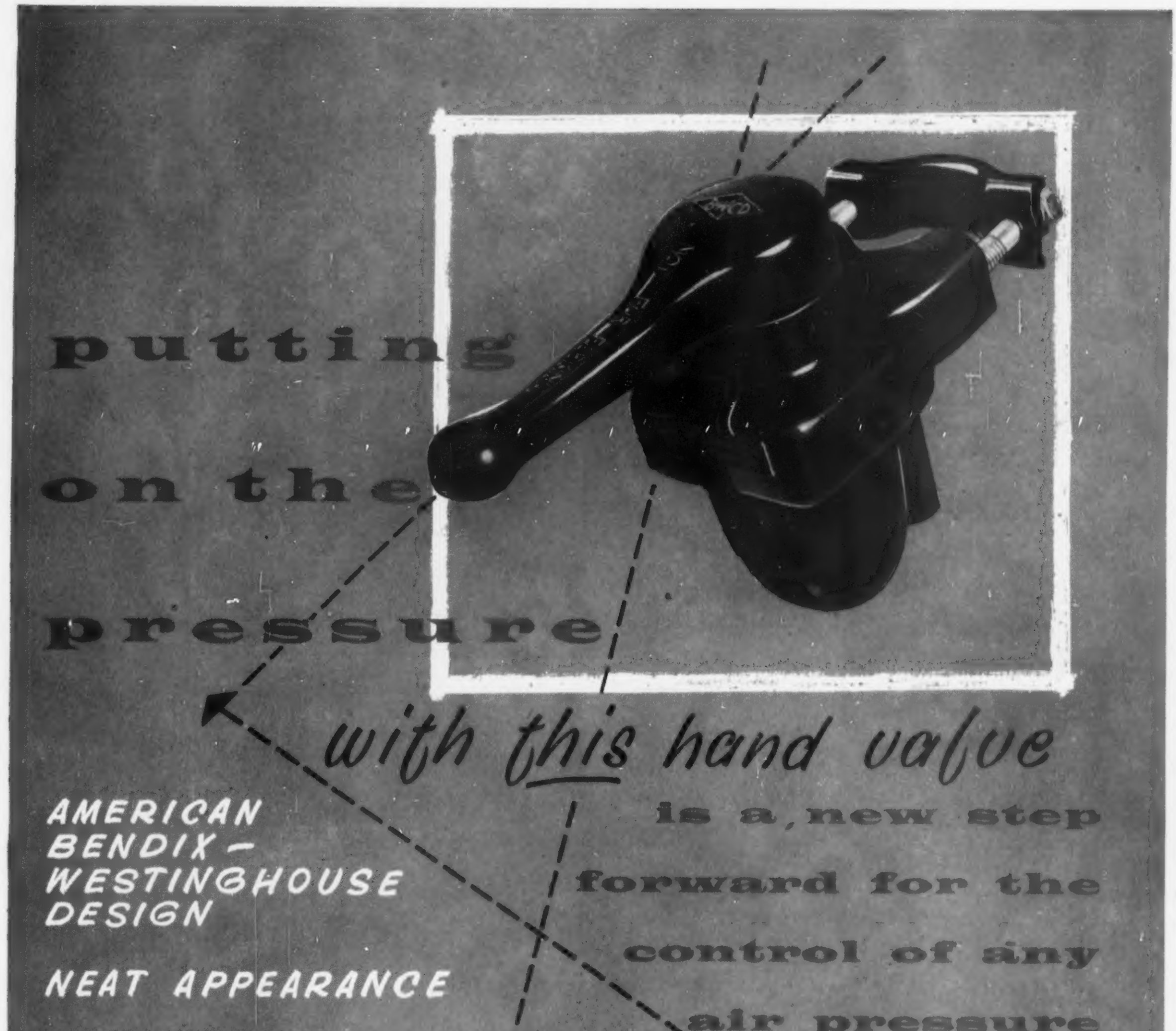
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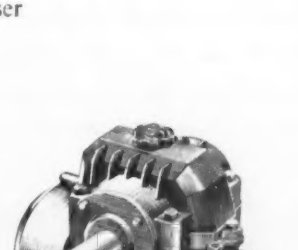

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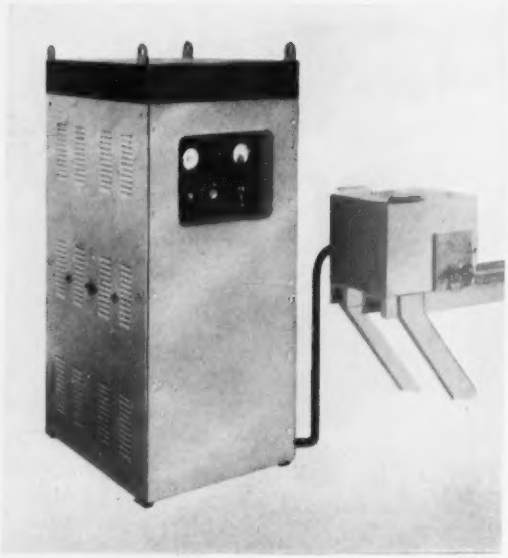

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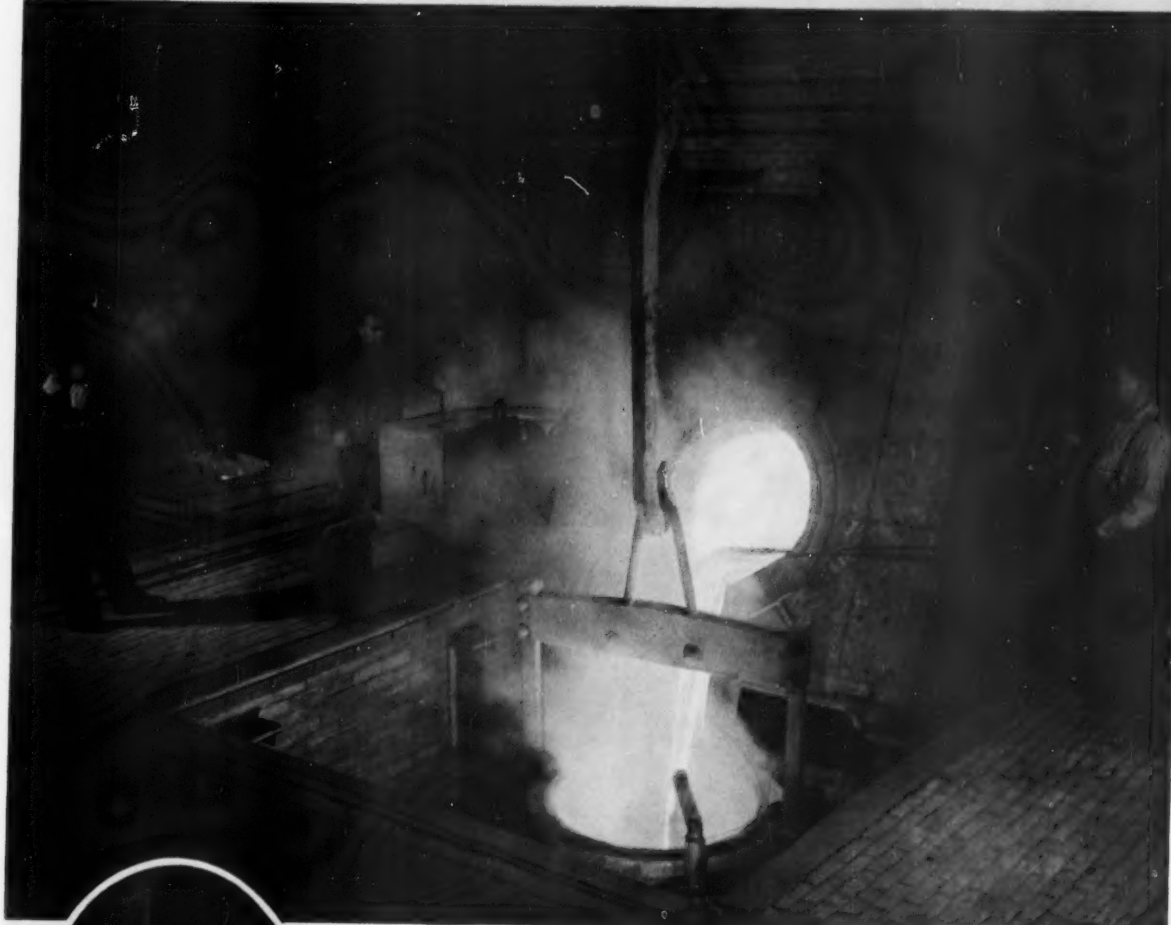

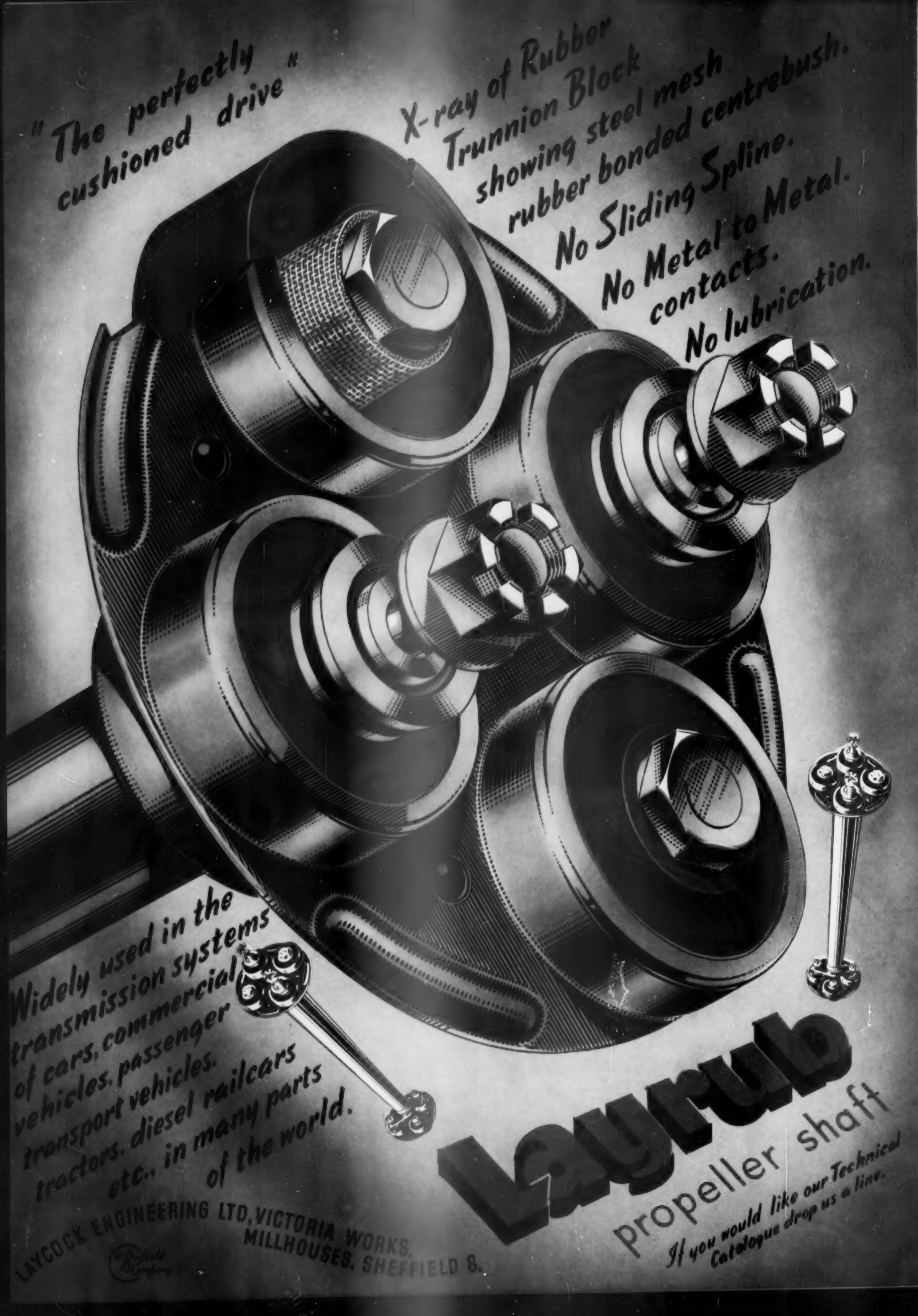
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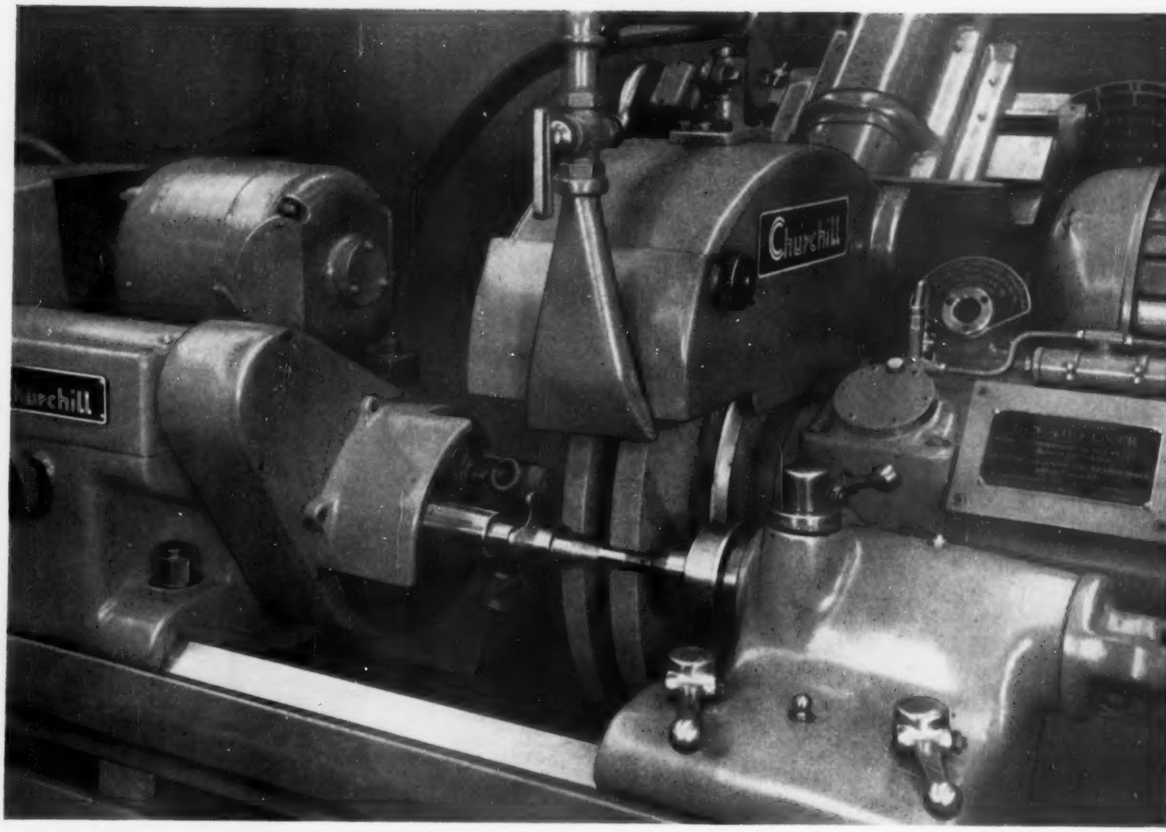
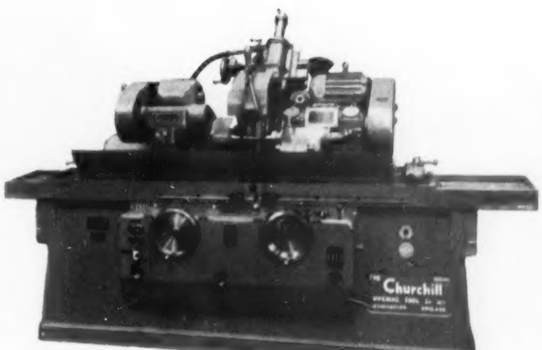


Churchill

THE CHURCHILL Model 'BW' 10″ swing Plain Grinding Machine has a reputation for accuracy and high output. The machine shown here is equipped with twin grinding wheels for grinding two diameters simultaneously. The wheel forming device is mounted permanently on the back of the wheelhead and enables the wheels to be trued in correct relationship to the finished work diameters.

The 'BW' Machine can be supplied as a non-automatic or fully automatic Plain Grinder.



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Cross slide feed is continuous, with automatic change-over from rough to finish grinding, and automatic truing of the grinding wheel.
This is the UHM P.195 which is fitted with a Face Grinding Head, shown lowered so that the facing wheel is in the grinding position. Internal grinding capacity $\frac{3}{4}$" to 8" diameter. There are other U.V.A. Internal Grinding Machines for grinding bores from $\frac{1}{2}$" to 18" dia.
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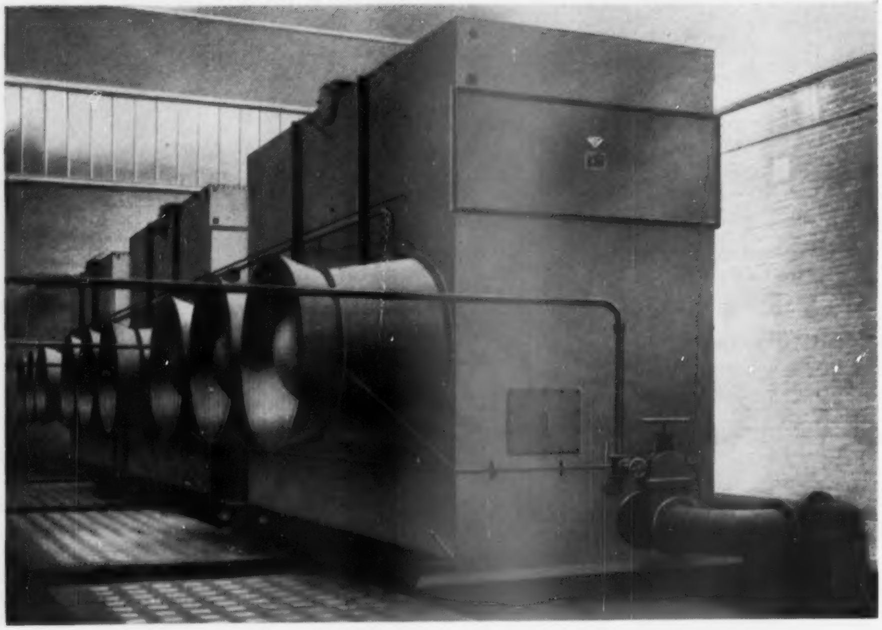

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for normal and special irons

For Normal and Special Cast Irons, standard G.W.B.—A. Tagliaferri Furnaces give an output of 140 to 2,000 lbs. per hour. Other data of their performance in relation to cast irons are shown in the panel alongside.

The furnace can melt cold charges or be fed with molten metal previously melted in a cupola. In either case, alloying additions can be made to produce high duty irons, the mechanical motion of the bath ensuring complete alloying and homogeneity of product. The furnaces are particularly suitable for thin wall castings (automobile cylinder blocks, etc.).

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1 Initial starting without using hot metal.

2 Positive metal circulation without turbulence gives maximum refractory life.

3 The refractory lined casing is easily exchanged with the relined spare without removing electrical connections or inductors.

4 Clear indication is given when end of lining life is approaching.

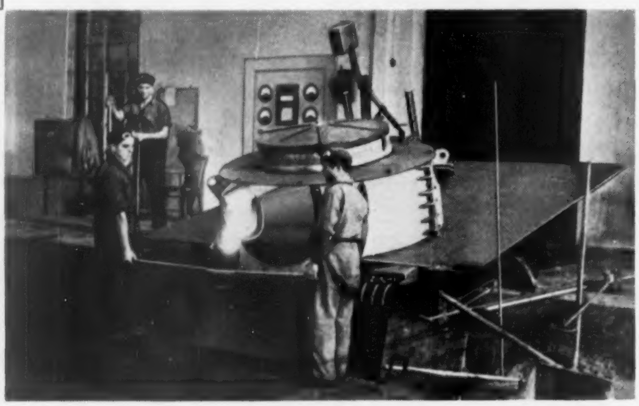
5 Refractory lining of melting duct can be repaired without dismantling the furnace.

Full details of standard units suitable for all normal foundry requirements may be had on request.

Melting Furnaces for Normal and Special Cast Irons

TYPE	G 50	G 100	G 150	G 200	G 300	G 400	G 500
CAPACITY LBS. TOTAL	550	990	1760	3300	4400	6600	11000
USEFUL	440	770	1320	2200	3300	4840	8800
RATING kW	37.5	75	110	150	225	300	450
kVA	50	100	150	200	300	400	550
OUTPUT LBS. PER HR.	143	286	396	660	990	1430	1980
CONSUMPTION kWH/TON	558	558	538	508	478	467	437
TILTING METHOD	Hand	Hand	Hydr	Hydr	Hydr	Hydr	Hydr
HOW CONNECTED	1 Phase	1 Phase	1 Phase	3/2 Phase	3/2 Phase	3/2 Phase	3 Phase

The above figures which are for cold metal charges are not binding and will vary according to the product and the quality of the metal charged.



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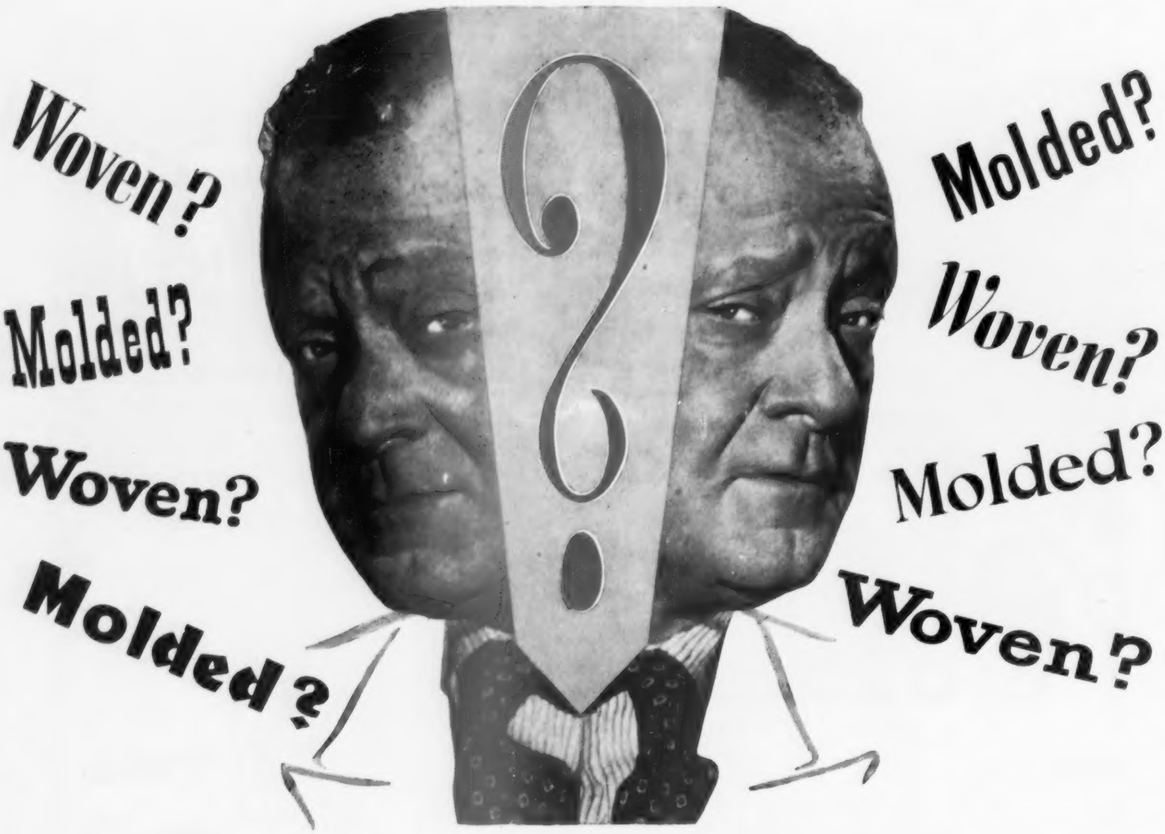
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Molded?
Woven?
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Molded?
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Woven?

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ROLLER JOURNALS

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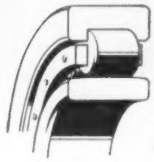
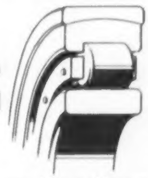
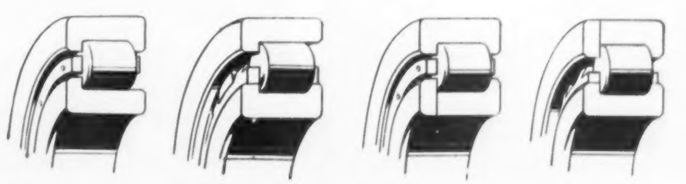


The normal accepted type has a parallel outer ring and shouldered inner ring as shown above, but where conditions demand, bearings are made with a parallel inner ring and shoulders on the outer ring.



When it is necessary for a roller bearing to accommodate misalignment, bearings are supplied fitted with a spherical shell.

Roller bearings are usually required for journal loading only, but under approved conditions bearings made with additional shoulders or sideplates can be employed for location duties. Some of these are here illustrated.



- ★★ ROLLER JOURNALS
- ★ BALL JOURNALS
- ★ DOUBLE PURPOSE BEARINGS
- ★ THRUST BEARINGS
- ★ LINESHAFT BEARINGS
- ★ NEEDLE ROLLER BEARINGS
- ★ AIRCRAFT BEARINGS

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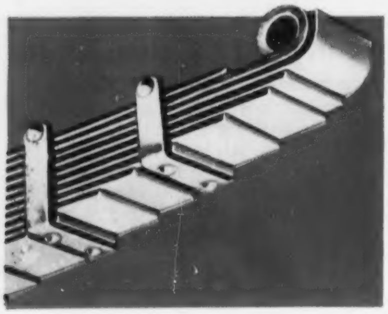

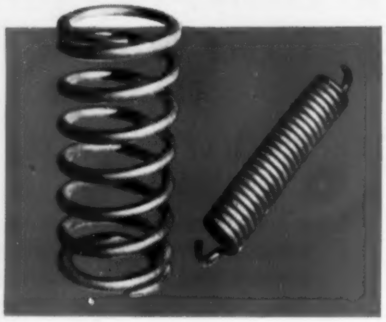

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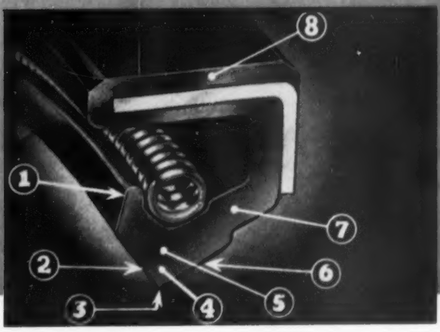

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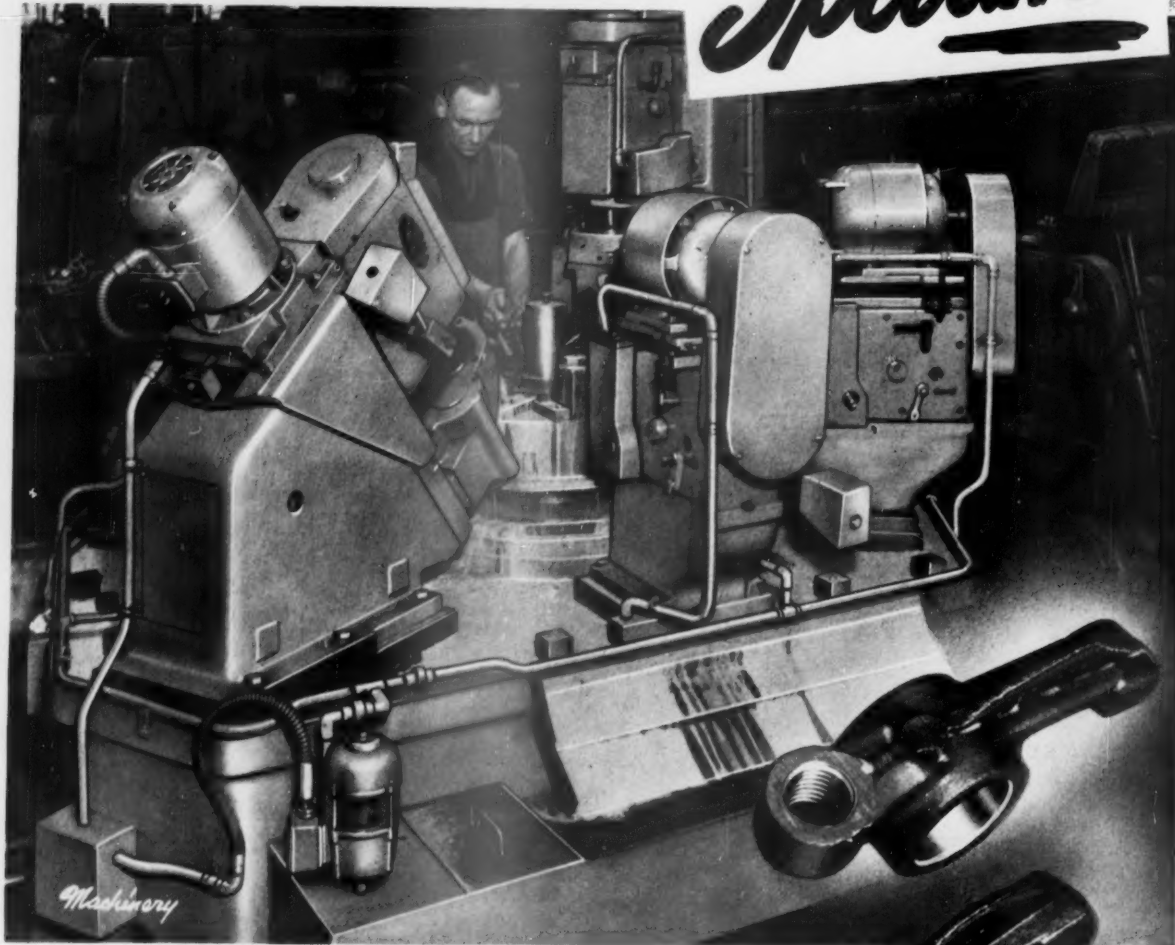

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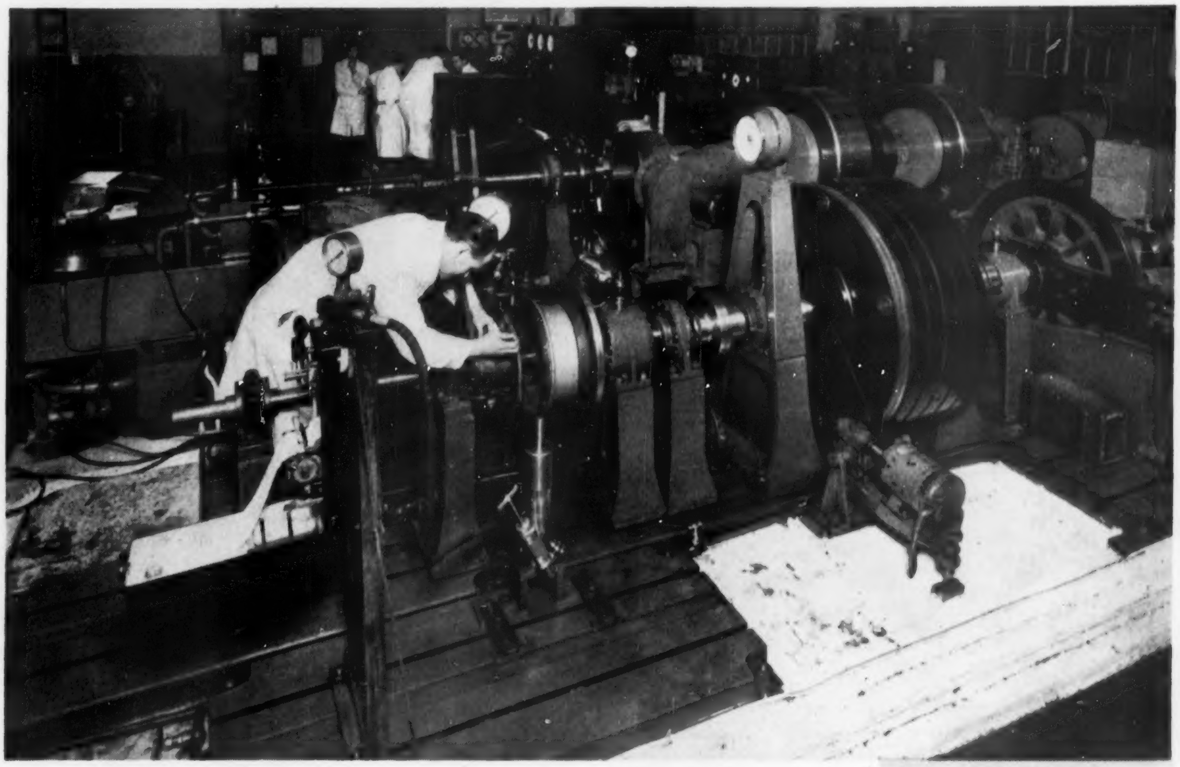

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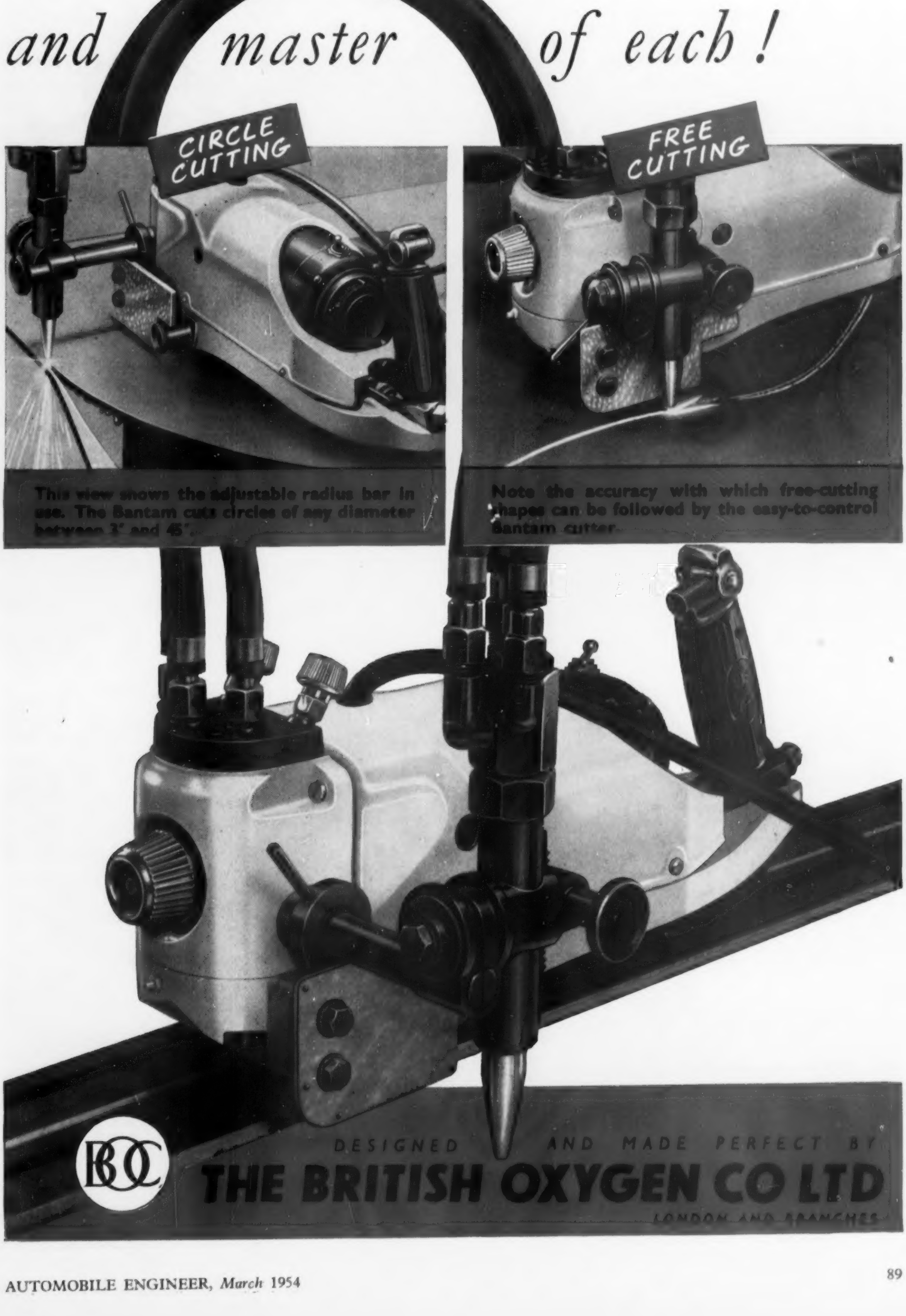
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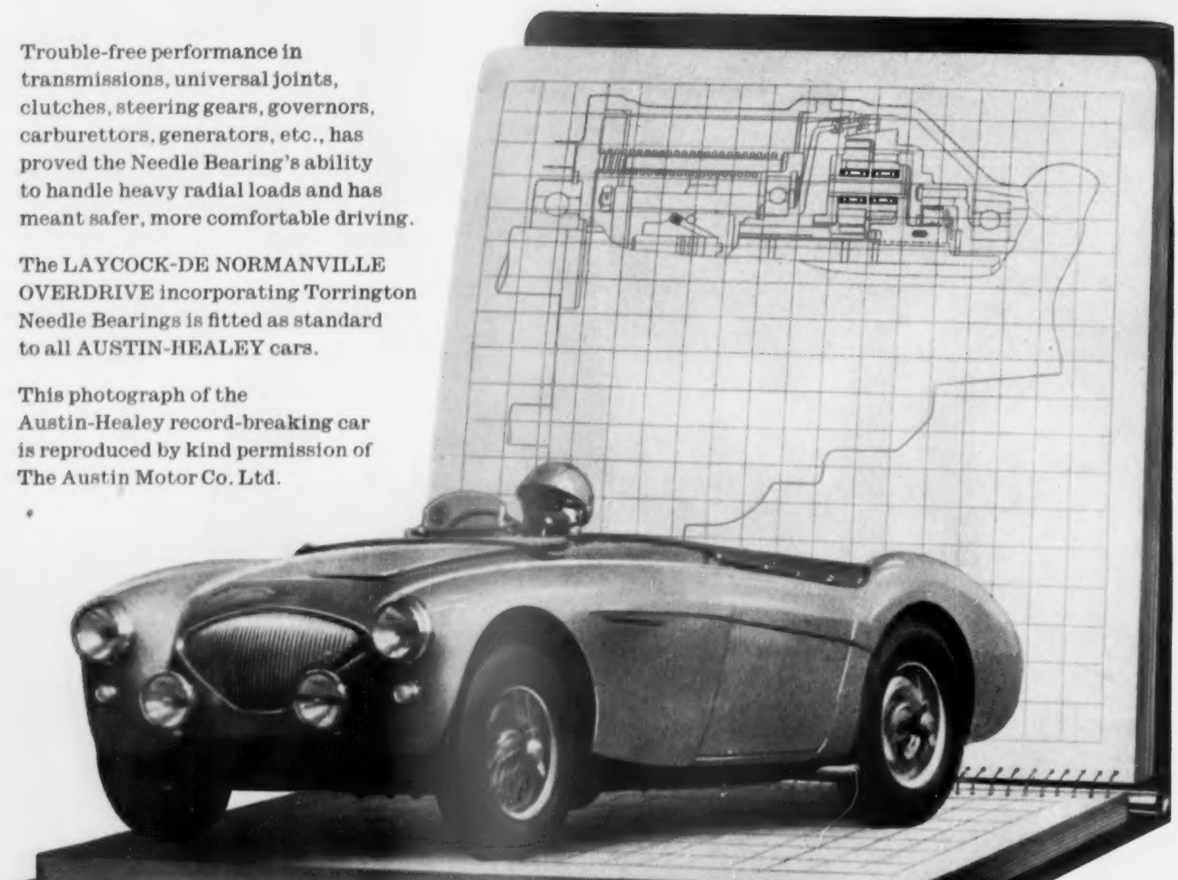

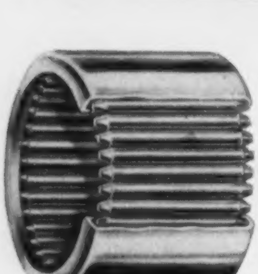

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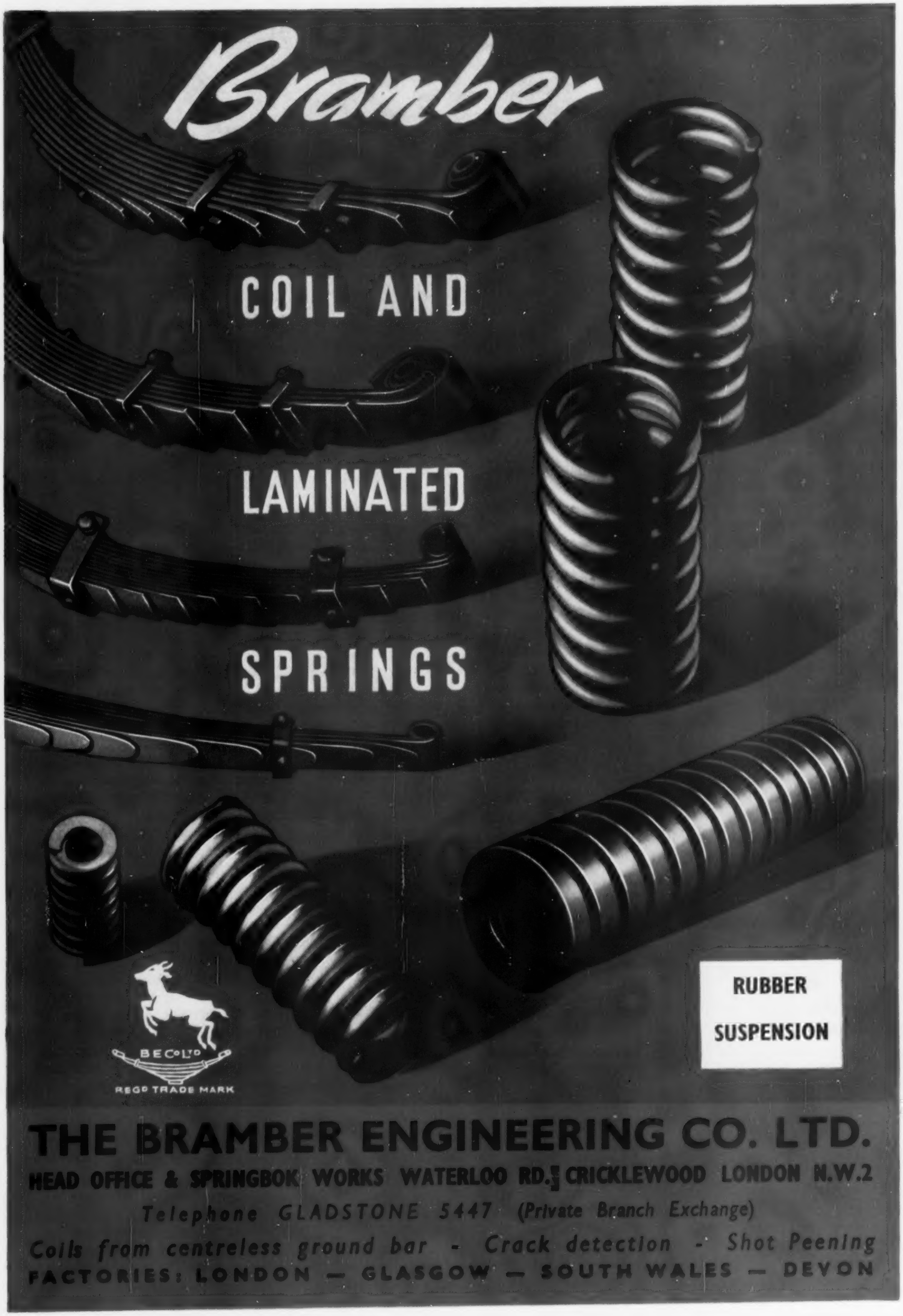
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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Ill-Informed Criticism

ALTHOUGH British automobile manufacturers must by now be inured to criticism from many different quarters, they will doubtless be greatly surprised, but not perturbed by the most recent strictures upon the manner in which they conduct their affairs. The surprise will be occasioned by both the source, the industrial experts—whoever they may be—of the United Nations and by the content of the strictures. Briefly, these experts say, "there is no sign of a fresh approach to the design of a popular car, particularly one which takes advantage of weight reduction offered by careful design and the maximum use of light alloys." They continue, "because many British manufacturers have again entered the small car market, it is certain that an output of one model that is really economical is out of the question. This is in direct contrast with Western Germany."

Obviously, the criticisms can be aimed only at the "Big Six". It is therefore germane to recapitulate briefly their post-war record. These organizations have in the past four or five years made capital investments in the order of £50,000,000 in modernizing their plants and in tooling for new models. This in itself proves nothing, but it can be claimed that the results have been outstanding. In the automobile industry productivity has increased much more than the national average for engineering, and indeed, for any major industry. At the same time, earnings of workers in the industry are much higher than the average. Yet, despite the vast capital investment and the high level of wages, the percentage increase in car prices in comparison with pre-war levels is much less than the average. The immense contribution the industry has made to the export trade of this country does not need to be emphasized. This is hardly the record of an industry that needs advice from "industrial experts".

The "popular" car

Unfortunately, but not surprisingly, there is a certain imprecision in the terminology of the strictures. What, for example, is the meaning of a "popular" car? In the context it is probably meant to connote a car from which all but the basic necessities are excluded; that is, the type of car that in this country is generally referred to as an austerity model. Post-war experience, and more particularly the experience of the past year, proves that the idea that there is a large demand for a thoroughly austerity car is mistaken. Such a car would not be acceptable to a large market in this country and at most would meet with acceptance in only one or two overseas markets, and then only in the face of competition from established Continental vehicles. There is a certain minimum standard of refinement that must be maintained if sales are to be of an order to warrant the expense of tooling for large quantity production.

Nor is there much point in the statement "weight reduction offered by careful design and the maximum use of light alloys". That is a generalization that may sound well when it is uttered but is in fact almost meaningless. Weight reduction is a means, not an end, and it must be considered in relation to other factors. Reduced weight will, of course, lead to reduced fuel costs, but it must also be remembered that at present prices a wider use of light alloys would increase the cost of a vehicle.

Economic output

Nor is there any reason for accepting the experts' dictum that "it is certain that an output of any one model that will be really economical is out of the question". Experience has shown that it is economically dangerous for a British manufacturer engaged in quantity production to rely on a single model, and this indeed applies to Continental manufacturers also. For example, Fiat, with a much smaller home market than the British, produces at least five models.

It is not easy to understand why from time to time Volkswagen practice is held up as a pattern for the British automobile industry to follow. The output of the Volkswagen car may be greater than that of any one British model, but this in itself does not establish that British outputs are not economic in quantity. The ultimate criterion must be the value the customer receives for the purchase price he pays, and on this basis the British automobile industry does not compare unfavourably with the Volkswagen organization.

Automatic Machine Control

IT is amazing how frequently developments made for a certain specific purpose prove to have very much wider applications. A recent example of this is a new method, suggested by Sir Thomas Merton and being developed by the National Physical Laboratory, for making diffraction gratings. Although they are primarily intended as spectroscopes for analysing radiation, the "Merton-N.P.L." gratings are also applicable to industrial

processes. They can, for example, be used for precise measurement of length and the control of machine tools. For these purposes long, accurate transparent gratings are needed. They can be produced cheaply by the new process.

If two of these gratings are placed one upon the other with their linings inclined at a small angle, dark lines or fringes are produced. When one of the gratings is kept at rest and the other is moved very slowly in the direction of the fringes, the fringes also move but at a greatly magnified rate. The number which passes across the field of view can readily be counted by a photocell or electronic counter. This method lends itself to very accurate and rapid measurement.

An industrial firm is collaborating with the Light Division of the N.P.L. in applying this technique to the control of machine tools such as jig borers. They are also working out further applications for the automatic control of machines. When this work is completed it will be possible for elaborate machining operations to be performed from instructions stored up in a tape machine. The truly automatic factory may be nearer than we think.

American Engines

An outstanding feature of American passenger cars in 1952 and 1953 was the installation of engines with greatly increased maximum brake horse power. This was general throughout the whole of the automobile industry in the United States of America. In the main the greater b.h.p. was obtained by developing engines that had a higher b.h.p/displacement ratio rather than by increasing engine capacity. Generally, the increased b.h.p. was achieved at the expense of an inferior torque curve, poorer general performance in the lower speed range and increased fuel consumption.

An analysis of information available for passenger car engines for 1954 shows that there is no longer uniformity of thought throughout the American industry on the question of increased maximum b.h.p. However, some organizations are still following the previous trend, notably Chrysler with an engine that develops 235 b.h.p., an increase of 31 per cent over the power output of the 1953 engine.

Several makers have adopted a completely different policy. For example, the Lincoln engine for 1954 has the same displacement, 317·5 in^3, and the same maximum b.h.p., 205, as the 1953 engine. The new engine has been modified to give an improved torque curve and better acceleration in the lower speed range. Packard and Oldsmobile are installing engines of greater maximum b.h.p., but the increased horse power has, in each case, been obtained by increasing the engine displacement without any attempt to increase the output/displacement ratio. Both these engines have flatter torque curves than had their predecessors. They are also said to be much more economical on fuel, although this improvement is at least in part due to modifications of the axle ratios.

There is a distinct trend towards a wider use of overhead valve V-8 engines. An analysis of 23 engines for 1954 passenger cars shows that thirteen have OHV V-8, three OHV V-6, four in-line 8 and three in-line 6 engines. Cars that have V-8 engines for the first time are the Ford, the Mercury and the Buick special. In 1955 it is probable that Chevrolet and Pontiac will also have V-8 power units, and some observers consider that this type of engine will, within a few years, be the standard type of power unit for American passenger cars.

Compression ratios

Compression ratios continue to rise. The ratios employed vary from 7·2:1 to 8·7:1. An analysis shows that of 23 engines eight have a compression ratio of 8:1 or more. Obviously the fears that suitable fuels for high compression ratios would not be commercially available must have been allayed. In fact, fuels of from 92 to 97 octane rating are now available at American service stations. It is therefore not improbable that even higher ratios, perhaps approaching 10:1, may be expected in the near future.

One interesting inference may be drawn from this tendency towards higher and higher compression ratios. High octane fuels are not commercially available to any great extent in the export markets for passenger cars; nor are there any signs that they will be for some considerable time to come. Therefore, high compression engines will not be suitable for use in such markets, and the inference must be that for some time to come the American automobile industry is prepared to forgo much of the export trade that it held in pre-war days.

CONTENTS

THE CITROËN 2CV

Part I. A Critical Review of the Engine Design

INTEREST in the Citroën 2CV has been intensified in this country by the announcement that it is to be assembled at the Citroën factory at Slough. Despite the fact that this is one of the most unconventional cars produced in large quantities since the war, it is also one of the most successful in its country of origin. Its success is in part due to the compact engine design, which has made possible the adoption of a front-wheel-drive arrangement. This not only allows exceptionally easy access to the whole of the engine and transmission system, but also leaves the maximum amount of space free in the body.

As a result, the cost and overall dimensions of the vehicle have been kept to a minimum. However, the economies effected by the use of the front-wheel-drive layout are to some extent offset by the fact that one additional sliding joint, two extra universal joints and an extra pair of rolling element bearings are required, as compared with the more conventional rear drive arrangement.

Another attractive feature of this vehicle is its low fuel consumption in terms of miles per gallon. This has been obtained by employing an engine of small capacity, 375 cm³, and by keeping the weight of the vehicle to a minimum. The power output of the unit is 9 b.h.p., which represents 24 b.h.p/litre, and is low even for a car of such a small size. From the figure of 1,100 lb, for the kerb weight, given in the specification panel, it can be seen that the power:weight ratio is 18·35 b.h.p/ton.

Particularly low rates of wear of cylinder bores, as well as of other components, are characteristic of this unit, and they are probably partly due to its low rating. It is obvious that if certain changes were made, notably to the induction system and crankshaft, a much higher output could be developed. Features such as rapid warm-up when starting from cold and relative freedom from cylinder bore distortion, which are inherent in horizontally opposed, air-cooled, twin-cylinder layouts, undoubtedly also help to reduce the rate of bore wear.

Other details of this engine are given in the accompanying specification panel. The maximum b.m.e.p. and torque are 110 lb/in² and 16 lb-ft respectively, and are developed at 1,800 r.p.m. A bore:stroke ratio of 1:1 has been adopted and the connecting rod length:stroke ratio is approximately 1·93:1. At the engine speed at which maximum b.h.p. is developed, the mean piston speed is 1,430 ft/min. This speed is very low and although, as has already been stated, it is not the only factor that influences cylinder wear, it undoubtedly tends to prolong the periods between rebores. The output per square inch piston area is 0·95 b.h.p.

SPECIFICATION

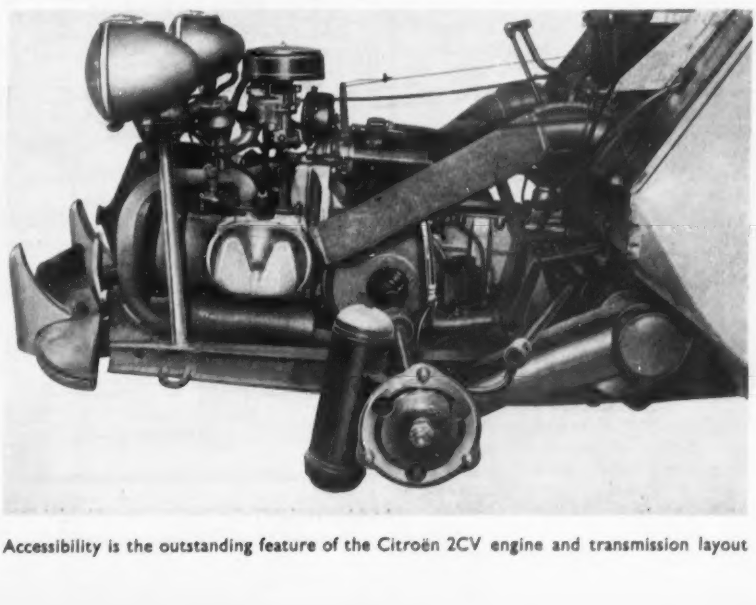
ENGINE: Two cylinders, horizontally opposed. Bore and stroke 62 mm × 62 mm. Swept volume 375 cm³. Four-stroke. Maximum b.h.p. 9 at 3,500 r.p.m. Maximum b.m.e.p. and torque respectively 110 lb/in² and 16·6 lb-ft, at 1,800 r.p.m. Compression ratio 6·2:1. Two-bearing, two-throw crankshaft, built up from five pieces. Push rod operated, overhead valves. Solex carburettor and Guiot fuel lift pump. Fuel tank capacity 4·4 gallons.

A problem that has to be faced with a front-wheel-drive layout is that the engine mountings must be designed to react the final drive torque as well as to isolate the frame from the normal vibrations of the engine about its longitudinal axis of oscillation. Moreover, with horizontally opposed, twin-cylinder engines, rocking couples about a vertical axis must be catered for. The final drive torque, of course, acts about a horizontal axis normal to the longitudinal axis of the unit.

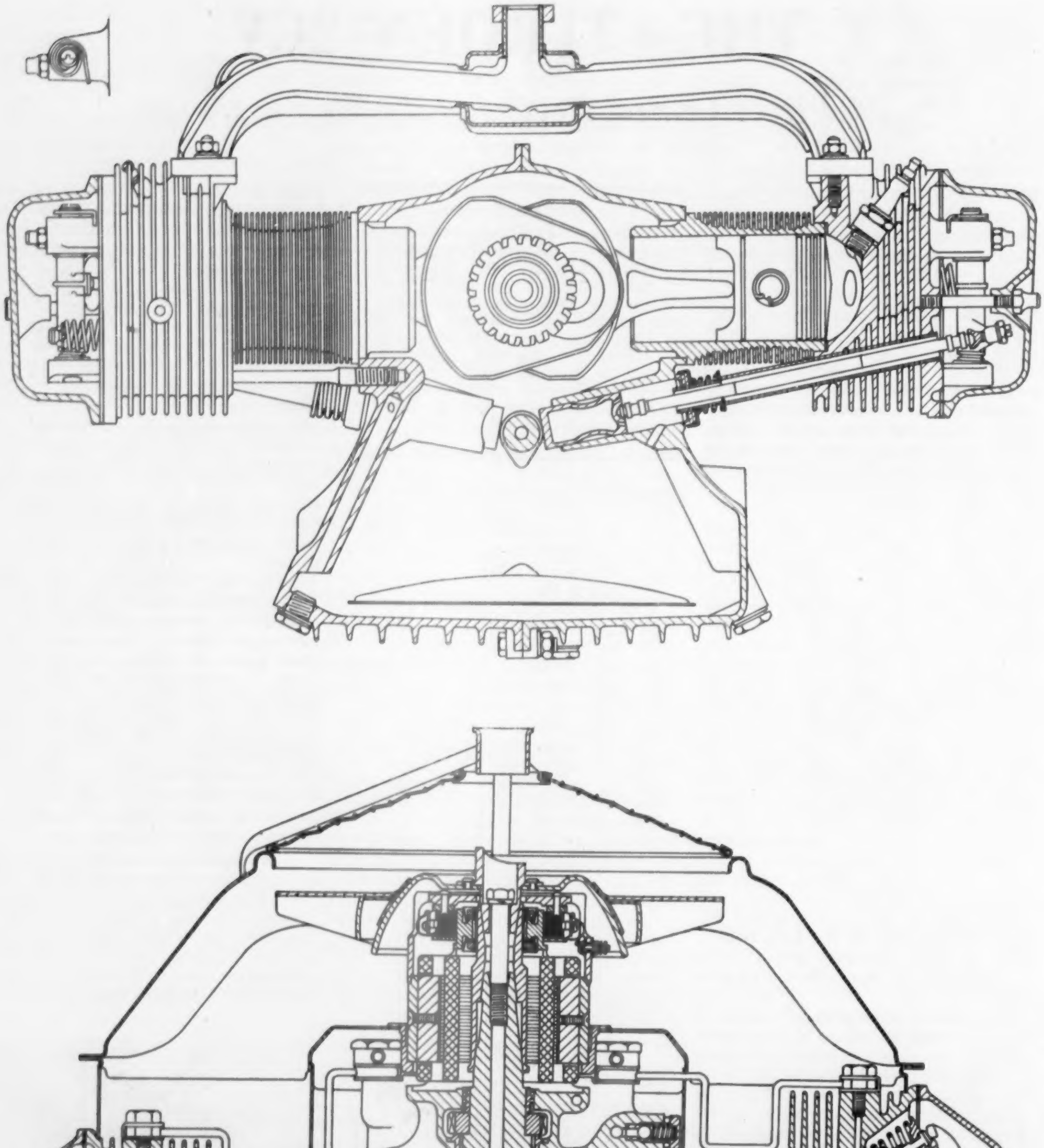
To meet these requirements, a V-arrangement has been adopted for the layout of the front mountings, but instead of the usual single-sandwich rubber unit, a double-sandwich type mounting is fitted on each side. The centre plate and the rubber above and below it are enclosed in a short rectangular section, tubular housing which is bolted to a bracket on the front cross member of the frame. On each side, the centre plate is bent upwards and then flanged outwards. A stud is welded to the top face of each flange, and these studs are used to secure the unit to a right-angle bracket bolted to the front wall of the crankcase. The up-turned ends of the centre plate act as limit-stops to prevent the amplitude of oscillation of the engine about its longitudinal axis from becoming very large. However, it is doubtful whether they are ever called upon to perform this function. The reaction at the front to the final drive torque is taken by rubber in compression either above or below the centre plate according to the direction of the drive.

At the rear, the mounting is a rubber bush, the longitudinal axis of which appears to be approximately on the axis of oscillation of the engine. This bush is not of circular section, but is oval, the major axis of the oval being horizontal. In this way, a relatively large area of rubber is available to take the compression loads imposed on it by the final drive torque. At the same time, the horizontal rocking couples are reacted by the relatively small areas of rubber in compression at the ends of the major axis of the unit.

The mounting is carried by a swan-neck fitting welded on top of the cross tube supporting the front suspension pivots. The end of this fitting projects into the rear of the central steel tube of the mounting bush, where it is secured by two 7 mm diameter set bolts passed from the front through a vertical



Accessibility is the outstanding feature of the Citroën 2CV engine and transmission layout

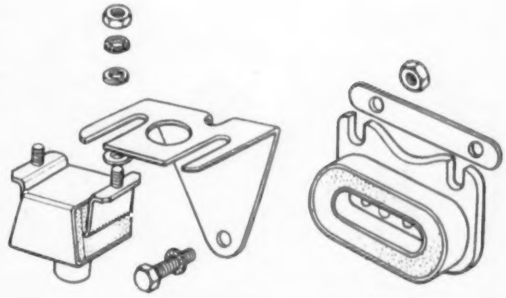


ARRANGEMENT OF THE CITROËN 2CV ENGINE
Bore and stroke 62 mm × 62 mm. Swept volume 375 cm³

plate welded in the tube. When the engine is installed, it is lowered so that two bolts, screwed into the rear face of the turret housing for the striker lever of the gear shift control, rest in U-shaped slots cut in the top edge of a vertical plate welded to the steel outer tube of the rear mounting bush. The bolts are then tightened against the plate.

An integral crankcase and sump is employed. It is an aluminium alloy casting divided in a vertical, longitudinal plane, and the halves are dowel located. At the rear, four 10 mm diameter studs are carried to secure the transmission casing to the crankcase. Two housings are cored in the front end of the crankcase. The inner one, the base of which is open to the sump, completely encloses the driving gear on the crankshaft extension and the half-speed wheel. Behind the driving gear is the front bearing of the crankshaft and in front of it is a ring pressed on to the shaft. This ring has an oil return scroll machined on its periphery; it is housed in a bore in the front wall of the crankcase.

In front of this is the dynamo, the hub of which is on a taper on the end of the crankshaft. A forward extension of the hub carries the blower rotor, and is internally tapered to receive a tapered plug pulled into it by a bolt screwed into the front end of the crankshaft Dogs for the starter handle are formed on the plug. The shell of the dynamo, in which the field coils are mounted, is bolted to the crankcase. With this arrangement, there are no bearings in the dynamo and therefore reliability may be improved and cost reduced. However, the unit has to be designed specifically for the engine and this tends to offset the advantage of cost reduction. Moreover, in some engine designs, because of crankshaft deflections, the adoption of such a layout might present some difficulties.



The front and rear engine mountings react the final drive torque in addition to isolating the frame from engine vibrations

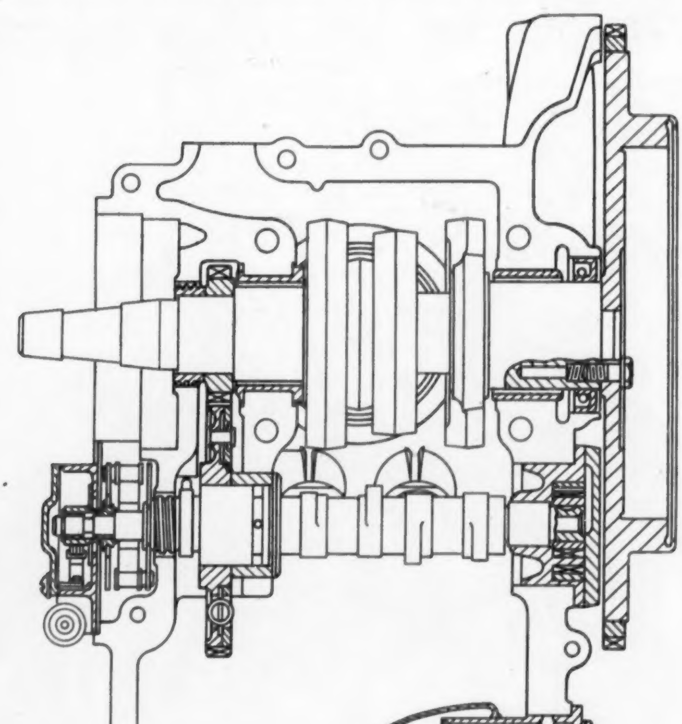
Crankshaft, connecting rods and pistons

The crankshaft and connecting rods, together with the front journal bearing and the pressed-on helical gear and oil return scroll ring at the front end, are fitted to the engine as a complete assembly during manufacture, and the components cannot be obtained separately as service spares. This is because the crankshaft is built up from five pieces, and the connecting rods, together with the big end bearing bushes, are assembled on to the crank pins before they are pressed into the webs. The five pieces are: the front and rear ends complete with webs, the two crank pins, the ends of which are shouldered to locate them axially in the webs, and the oval centre web. In an engine of much higher power rating, the press fit alone would hardly be adequate to resist the torque loading of the shaft, but apparently movement of the components relative to one another is not experienced in this application.

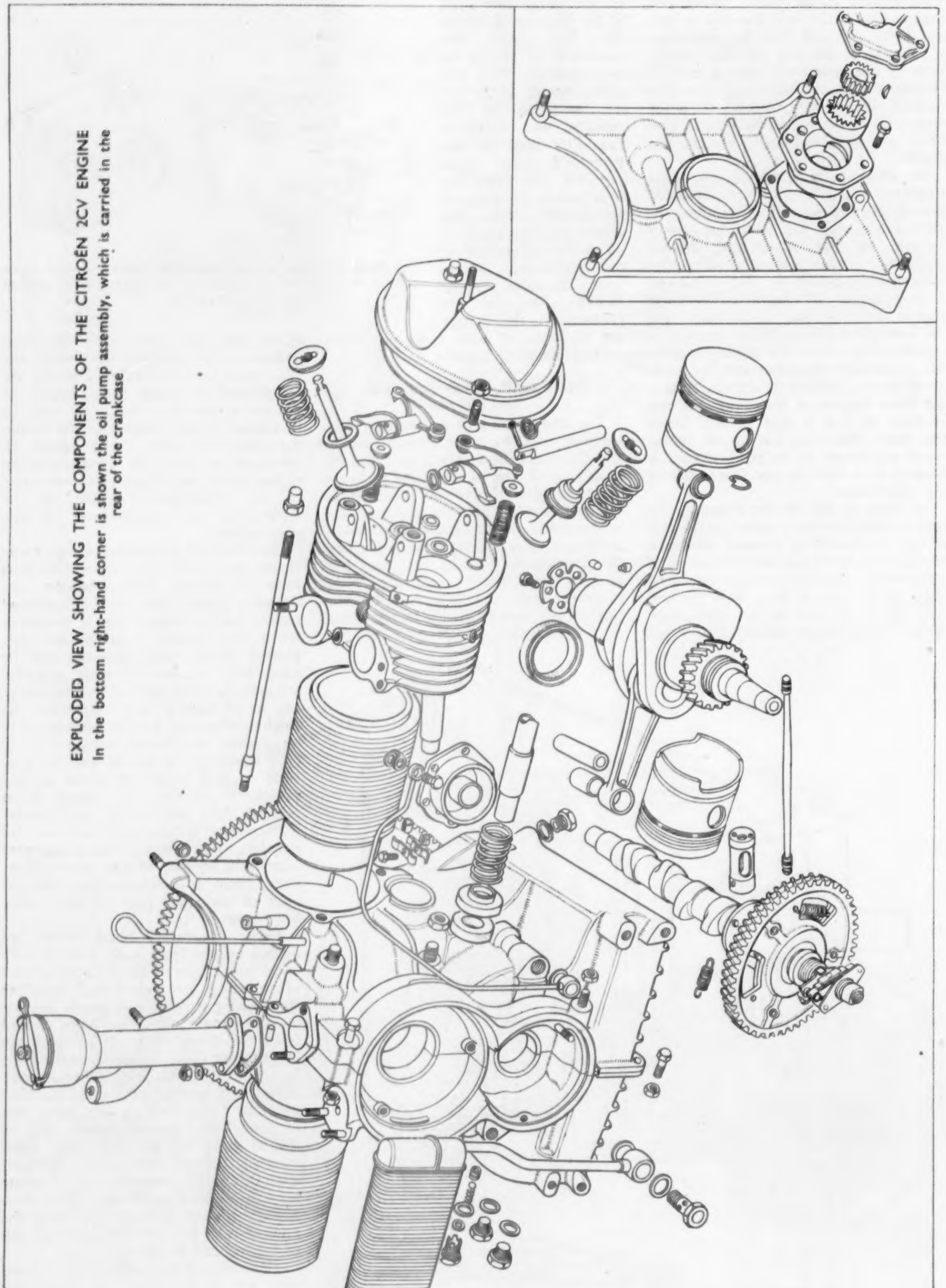
This method of manufacturing crankshafts may well solve more problems than it creates. For instance, the separate crank pins can be hardened readily to the degree that is necessary when lead bronze bearings are employed. Then, plain bearings can be used and, because of their superior strength as compared with white metal, can be of such a short length that no further advantage in this respect could be gained by fitting rolling-element type bearings. It would also be possible to mill small serrations on the periphery of the pin where it is pressed into the softer crank webs. This might reduce any tendency for rotational movement of one component relative to another to take place. However, there is no indication that the ends of the crank pins on this engine are serrated.

Large diameter, blind holes are drilled axially from each end of both crank pins, presumably with the object of reducing the weight and therefore the rocking couples and inertia loading on the crankshaft. The holes drilled from the outer ends of the pins are longer than those drilled from the ends in the centre web and they are each closed by a pressed-in, cupped blanking plug. A radial hole in the outer end of each pin communicates with a drilled hole in the web and main journal: through these holes, lubricating oil is taken from the main bearing into the hollow crank pin. Two more radial drillings, mid-way between the ends of each pin, feed the lubricant out to the big end bearings.

The axes of the main journal bearings are in the plane on which the crankcase is divided. Axial location of the shaft is effected by the front bearing, which is flanged at each end. The dimensions of the rear bearing are 48 mm inside diameter by 56 mm out-



Vertical, longitudinal section of the engine. The rear bearing of the camshaft is formed by the oil pump body



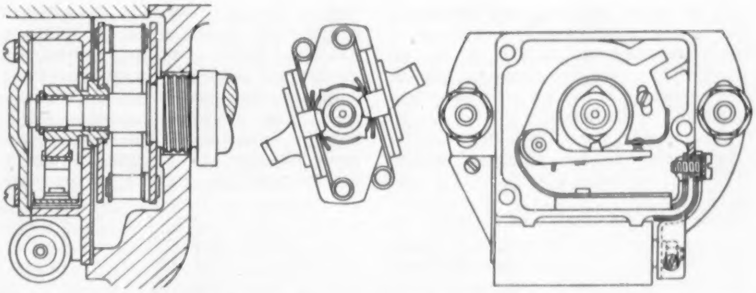
EXPLODED VIEW SHOWING THE COMPONENTS OF THE CITROËN 2CV ENGINE

In the bottom right-hand corner is shown the oil pump assembly, which is carried in the rear of the crankcase

side diameter by 32·5 mm long. Both bearings are of white-metal lined bronze and are located by shouldered dowels. The larger diameter portion of each dowel is, of course, in the bearing housing, and the smaller, 8 mm diameter, portion is in the bearing, so there is no possibility of the dowel contacting the journal and scoring it. By using white-metal lined bearings, a certain amount of misalignment of the shaft can be tolerated, and in this engine they also appear to be large enough to cater for considerable out-of-balance forces.

Both the front and rear ends of the crankshaft are counterbored to reduce weight. At the rear, an oil thrower ring is pressed-on and is located against a shoulder immediately behind the rear journal. This thrower works in a space cored in the crankcase. Immediately behind it is a lip-type oil seal, which is housed in the crankcase, and which bears on the tail end of the shaft. The flywheel is spigoted on to the end of the shaft, and secured by five 8 mm diameter set bolts and located by one 8 mm diameter dowel. It is 252 mm diameter over the periphery on which the starter ring gear is pressed. The five bolts are locked by a circular plate, the edges of which are turned up against the hexagon heads of the bolts. The ring gear, which has 107 teeth, is pressed on, and it meshes with a nine-toothed pinion.

The big end bearing housings are not split, but are of the ring type. With this arrangement, the bearing lengths can be kept to a minimum because there is no discontinuity in the housing, and also because no bolts have to be accommodated. A further reduction in length has been obtained by the use of copper lead bearing shells. As a result the offset of the cylinder axes, and therefore the rocking couples, are relatively small. Phosphor bronze bushes are fitted in the small ends of the I-section connecting rods. The dimensions of the bushes are: inside diameter 20 mm, outside diameter 22 mm, length 25 mm. Fully floating gudgeon pins are employed. They are retained by wire circlips in grooves in the piston bosses. The bearing length of the pin in each boss is approximately 13 mm.



The automatic advance-and-retard and contact breaker units are mounted on the front end of the camshaft

Aluminium alloy pistons with domed crowns are used. They are somewhat unusual in that at the thrust faces, the base of the skirt of each piston is extended downwards. This presumably is to increase the bearing area available to take the thrust. It may also be intended to improve piston cooling. However, many authorities consider that by far the greatest proportion of the heat flow to the cylinder walls takes place through the rings and not between the contacting faces of the piston and the bore.

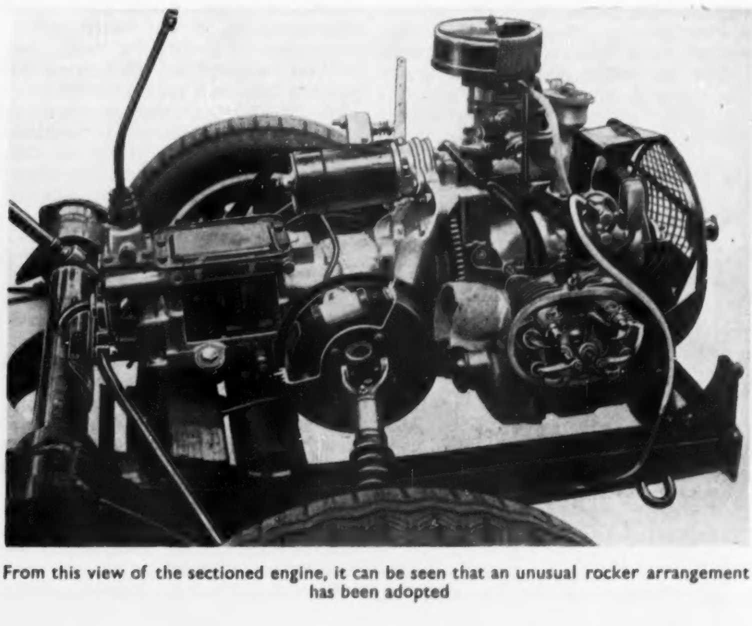
The pistons are of such a sturdy design that they appear to be capable of dissipating much greater heat flow than is likely to be experienced in an engine of such low rating. However, the precautions taken in this respect may well be justified in view of the fact that in a motor car powered by an air-cooled engine, the flow of air over the cylinders is not so great as in, for instance, a motor cycle. Three compression rings, each with a face width of 2 mm, and one oil control ring, with a face width of 3·5 mm, are fitted to each piston.

Timing gear, camshaft and valve gear

An unusual feature of this engine is that, so far as spares are concerned, the camshaft, its front bearing, the half speed wheel, the mechanical advance and retard mechanism and the contact breaker cam are supplied as a complete assembly. The contact breaker cam and automatic advance and retard mechanism are mounted on a forward extension of the camshaft. Behind these is machined an oil return scroll that works in a bore in the front wall of the crankcase. Immediately to the rear of this is the eccentric on which bears the push rod that actuates the fuel pump mounted on a boss on the right-hand side of the crankcase above the crankshaft gear.

The front journal is of such a diameter that the half speed wheel, which is pressed on to it, may be passed over the eccentric during assembly. A noteworthy feature of this half speed wheel is that it is in effect two wheels placed together on a common boss. These wheels are free to rotate independently before assembly to the engine, and after assembly they can still rotate relative to one another within the limits imposed by the meshing of the teeth. They are sprung in such a manner that the trailing flanks of the meshing teeth of one wheel are loaded against the leading flanks of those on the driven wheel, and the leading flanks on the other part of the half speed wheel are loaded against the trailing flanks of the driving wheel. Thus, there is no backlash and the timing gear runs relatively silently. The springs are of the coil type and are in tension. There are three of them housed in tangential slots in the wheel discs; one end of each is hooked into a hole in one disc and the other end is similarly attached to the other disc.

At the front of the camshaft, a bush-type bearing is employed. It is located between the half speed wheel and a flange on the rear end of the journal. Axial location of the whole assembly is effected by a stepped dowel, the small diameter portion of which registers in



From this view of the sectioned engine, it can be seen that an unusual rocker arrangement has been adopted

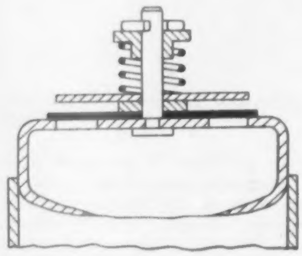
a hole in the bearing, and the larger end is in the housing. The rear end of the camshaft bears directly in the oil pump body, and the eccentric gear of the pump is keyed on to a rearward extension of the shaft.

The tappets are 24 mm diameter by 51 mm long and are of unusual design. They are of tubular construction and have hardened pads in their ends. Hardened spherical end fittings are carried at both ends of each of the push rods, which are 220 mm long. The rods are housed in tubes, the outer peripheries of which are shouldered at each end. The outer end of each tube is pressed and spigoted into the cylinder head, and then fixed by peening. Over the inner end is passed a washer, which is located against the shoulder on the tube and retains a coil spring. The inner end of this spring bears on an assembly, comprising a sealing ring and cupped retainer washer, round the end of the tube and presses it against the end face of the tappet housing bored in the crankcase.

The outer end of each push rod seats in the cupped end of a tappet adjusting screw on one end of the rocker arm. This arm is centrally pivoted and on its other end a cone is formed with its apex pointing inwards towards the head. The cone seats in a hole in a retainer washer for a light, coil-type return spring, which has the function of ensuring that the rocker at all times remains seated on the end of the push rod. On the other end of the boss of the rocker is another arm, extended in the same direction as the one with the cone end; it has a hardened end pad that bears on the valve in the usual manner.

There are two rocker shafts per cylinder. They are 12 mm diameter by 100 mm long. Their axes are vertical and they are carried between the two valves, which are set at an included angle of 70 deg in a horizontal plane. The pedestals that support the rocker shafts are cast integrally with the aluminium alloy head, and the upper pair form bosses through which two of the three cylinder holding down studs are passed. This arrangement is a good one from the point of view of accessibility to the holding down nuts.

During assembly, each rocker shaft is inserted from below. When the end of the shaft has been pushed a short distance through the hole in the lower pedestal, a compression coil spring, with a thrust washer at its lower end, is placed over it. Then the rocker, with another washer on its upper end, is inserted between the upper pedestal and the spring, and the shaft is pressed through until its upper end projects a short distance above the pedestal. Thus, the rocker and shaft oscillate together. The upper end of each shaft is stepped, so its cross section is of semi-circular form to clear the cylinder head holding down stud. A hole is drilled radially through it, and the inner end of a spiral spring is inserted in the hole. The outer end of this spring is bent at right angles to the plane of the spiral and is turned behind the rocker pedestal. On earlier models, a split pin was used in this hole to prevent the shaft and rocker assembly from bouncing downwards on the spring when the vehicle traversed rough terrain. Presumably the change was made because of a tendency under certain conditions for the pin to chafe

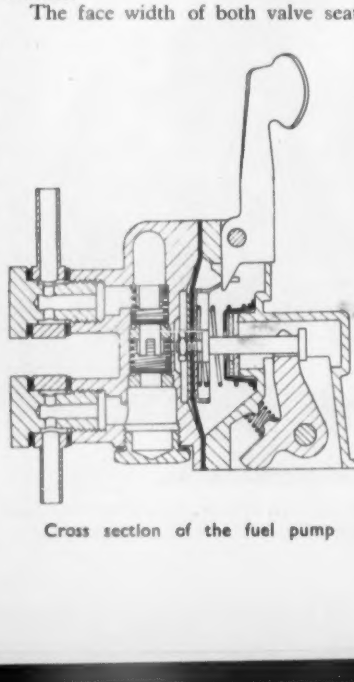


This breather valve is carried in a branch pipe on the oil filler tube

against the rocker pedestal. The larger area of contact between the spiral and the pedestal should be sufficient to prevent serious chafing.

Single springs are employed on the valves. Each has a free length of 38 mm and, under a load of 31-40 kg, its length is 24 mm, while under a load of 18-20 kg, it is 31 mm long. In the exhaust valve assembly, the lower end of the spring bears on a shroud which fits over the end of the valve guide to prevent it from being over-oiled, but in the inlet assembly the spring seats on a plain washer round the end of the guide. Each spring is retained by a shouldered washer, in which a hole is drilled eccentrically. A slot is cut in the centre of the washer to break out into the hole, and two flats are machined near the end of the valve stem. After the stem has been passed into the eccentric hole, it is slid into the slot in which it is retained by the end of the stem dropping into a counterbore in the centre of the washer.

The face width of both valve seats



Cross section of the fuel pump

is 0·9-1·5 mm. In the inlet ports, the included angle of the seats is 120 deg, and the face width is limited by cutting back the bore of the seat at an included angle of 26 deg. A 60 deg cut is effected in the bores of the exhaust valve seats to limit the width of the face, the included angle of which is 90 deg. The tappet clearances are 0·15 mm inlet and 0·20 mm exhaust.

Cylinder heads and barrels

Three cylinder-holding-down studs are employed and they are waisted to 8 mm diameter between the ends. They pass through the cylinder heads and are screwed into the crankcase. To prevent seepage of oil down the studs, cap nuts are used to hold down the cylinder head and barrel assembly. A single, 7 mm diameter stud is screwed into the centre of the cylinder head and a cap nut on it holds down the rocker cover. A joint washer is fitted between the head and the cover to prevent loss of oil. If this washer is not in good condition, or is badly fitted so that it leaks, it is possible in time to lose all the lubricating oil from the engine. It is difficult, however, to imagine such a large amount of lubricant being lost without its being noticed in time to prevent serious damage.

A spherical combustion chamber is incorporated in the head, and it has been necessary to machine a small recess in the crown of the piston to provide clearance for the inner edge of each valve during the overlap period at top dead centre. The head is spigoted on to the cast iron cylinder barrel, and a 1 mm thick copper gasket is fitted. This, together with the thickening of the barrel section adjacent to the cylinder head joint, ensures that there is an adequate path for the flow of heat from the head into the barrel, where it is dispersed by the cooling fins. Dissipation of heat from the cylinder head is, of course, a major problem in either water-cooled or air-cooled engines. The cylinder barrel section is also thickened near the flange that seats on the crankcase. A short spigot beneath the flange locates the cylinder. The remainder of the skirt, which projects into the crankcase, is machined to a smaller diameter. Thus, the area round the periphery of the barrel where close tolerances have to be maintained is kept to a minimum.

Induction system

In the cylinder head, the ports are turned upwards from the valve seats. The exhaust ports discharge vertically at the front and the inlets at the rear. A single joint washer bridges the two joint faces machined on the upper surface of the head, and the inlet and exhaust manifolds are pulled down on to it by three nuts and 7 mm diameter studs. One of these studs is screwed into the joint face in front of the exhaust port, another is screwed into the face behind the inlet port and the third is between the two, where the joint flanges abut.

Both the inlet and exhaust manifolds

are fabricated from steel tube and sheet metal. The inlet pipes from each side pass over the engine and are welded to the riser pipe at the centre. This junction is enclosed in a heater jacket, also fabricated from sheet steel, through which is passed the exhaust gas from the right-hand cylinder. On earlier models, a metal screen was interposed between this jacket and the carburettor mounted above it, but it is no longer provided, since it has been found unnecessary. The Solex carburettor is held down by two 7 mm diameter studs; a 6 mm thick, insulating washer is interposed between it and the riser pipe. A simple felt-element-type air cleaner is fitted on top of the carburettor.

A shouldered rubber sleeve connects a pipe from the air cleaner to a branch pipe on the oil filler tube assembly bolted on top of the crankcase. In the end of the branch pipe is a simple disc-type non-return valve to ensure that air can only pass out of the crankcase. This is an important feature because there is no oil filter in the lubrication system. Many authorities hold that it is unnecessary to incorporate a filter in lubrication systems provided foreign matter is prevented from entering the crankcase and sump.

Another interesting feature of the induction system is a small pipe that carries hot air, to prevent icing in extremely cold weather, to a point immediately above the entrance to the air tube in the centre of the diffuser. This pipe is clipped to the underside of the air cleaner. One end is extended horizontally through a hole in the side of the down pipe and bent at right angles over the diffuser. The other end is also bent downwards at right angles and is connected by a rubber sleeve to a vertical pipe welded to a small jacket clipped on to the exhaust pipe on the right-hand side. This jacket is a small pressing of semi-cylindrical form. One end, and the straight edges which are parallel to the axis of the pipe, are lipped inwards. The other end is left open so that air can be drawn into it over the hot exhaust pipe and then up the vertical tube into the carburettor intake.

An extremely simple control linkage connects the organ-pedal type throttle control to the lever on the stem of the butterfly valve. It consists of a single rod with one end bent at right angles and inserted into a rubber bush, 4 mm inside diameter by 8 mm outside diameter by 18 mm long, carried in an eye formed at the upper end of the pressed steel pedal. It is retained by a split pin and plain washer on the end of the rod. Screwed on to the other end of the rod is a ball socket by which it is attached to the lever on the carburettor. After adjustment for length has been made at this end, the setting is fixed by the usual lock nut arrangement.

Exhaust system

The exhaust from the right-hand cylinder is carried through a pipe, over the top of the engine to the heater box, which is round the junction between the induction pipes and the riser. It then passes out through another pipe to the left-hand side, where it is joined by a branch from the other cylinder. The manifold pipe terminates a short distance in front of this junction. A spherical seating ring is welded on its end, and the belled end of the exhaust pipe is clamped to it by means of a diametrically split ring, of channel section. Two 7 mm diameter bolts hold together the halves of the ring. The exhaust pipe is carried forwards and then downwards and to the rear, to an expansion box under the left-hand cylinder. From this box, another pipe carries the exhaust into the silencer which is mounted transversely beneath the gearbox.

Lubrication system

Oil is drawn through a strainer in the base of the sump into a long, thimble fitting on which the strainer is mounted and thence through a radial hole in the fitting into a vertical passage drilled in the rear wall of the sump and crankcase casting. From this passage, it passes into the eccentric-gear type oil pump. The eccentric gear is keyed on to the rear end of the camshaft. It drives an internally toothed annulus which is free to rotate in the housing, and which meshes at the top of the gear. Below, a crescent-section separator vane fills the clearance between the tips of the teeth of the gear and the annulus. There are thirteen teeth on the gear and sixteen on the annulus.

The oil outlet channel is cored in the cover plate and directs the lubricant into the hollow camshaft. Radial holes in the front end of the camshaft pass the oil into a large section annular groove round its front journal. From this groove, the lubricant passes into a drilled passage in the end of which is a spring-loaded ball-type relief valve housed in the side of the crankcase. The pressure should be 2·9-3·2 kg/cm^2 with the engine running at 3,500 r.p.m., and the oil temperature 80 deg C. Adjustment of this valve can be effected by interposing different distance washers between the end of the spring and the cap nut in which it is housed.

From the passage containing the relief valve, the oil passes forwards to a banjo connection on the right-hand side of the front wall of the crankcase, and thence to the oil cooler mounted, above the crankshaft, between the blower and the crankcase. The outlet from the cooler passes to a banjo connection on the left-hand side of the engine, whence passages are drilled to the main journal bearings. Holes in the crankshaft carry the oil to the hollow crank pins, and radial drillings distribute it outwards to the big end bearings.

Small pipe-connections taken from each of the banjo unions serving the cooler distribute oil through small diameter pipes to the cylinder head. From the union in each head, drillings take the lubricant to an annular groove round the exhaust valve guide, whence it is passed through holes into the rocker chamber. The oil return to the sump is through the tubes that enclose the push rods. Air is distributed from the eight-bladed, pressed steel rotor to the oil cooler and to the cylinders through a pressed steel shroud ring. A wire-mesh stone guard is fitted over the air intake, and in its centre, supported by three stays, is a guide tube for the starter handle.

FIRE EXTINGUISHERS

AT a recent demonstration, the effectiveness of the Antifyre Pistole Extinguisher was amply demonstrated. Two types were shown, one, the Model III, has a range of 25 ft, whereas the other, a smaller one termed the Lifeboat Unit, Model IIIB, has a range of 12-15 ft. During the demonstration localized fires, in which about two pints of petrol, two pints of cellulose thinners and a liberal quantity of liquid tar were burning furiously, were completely extinguished in a fraction of a second by one shot from the pistole. Then a large quantity of petrol was sprayed on the ground to form a trail about 10-15 ft long and ignited. Again one single shot extinguished the blaze.

The barrel of the unit is formed by the cartridge, which is fitted in a bayonet socket on the butt. Thus, if the first cartridge fails to extinguish the fire, a second one can be placed in position, ready for firing, in a fraction of a second. The overall length of the Model III is 14 in and it weighs 3½ lb when charged. In the Model IIIB unit, the cartridge is only half the length of that of the standard one, but is claimed to be equally effective except that, as has already been mentioned, the range is less. When the trigger is pulled, the hammer strikes a percussion cap in the centre of the base of the cartridge. This detonates a small explosive charge, which projects a fine white powder at the fire. The discharge is initially of conical form and then mushrooms out to blanket and beat the fire. It is also said to de-oxygenate the air in the immediate vicinity of the blaze.

Another type of extinguisher produced by the same manufacturers is the Primex unit. This contains carbon-tetra-chloride under pressure. By turning the cap at the top of the cylinder, the fluid is released and sprays out of a jet in the side of the unit. The discharge of fluid can be stopped by screwing the cap down and the extinguisher can be used again.

EXHAUST BRAKING EQUIPMENT

A Range of Control Systems for the Parkwood-Oetiker Unit

IN recent years more effective and consistently reliable braking has been demanded to match the steadily improved performance of both goods and passenger vehicles. This has re-focused attention on exhaust braking, a method hitherto virtually disregarded in this country although it has been widely used on the Continent for more than a quarter of a century. The apparent neglect occurred as a consequence of our relatively well-graded road system, the absence of any extensive mountainous terrain and the adequacy of conventional wheel brakes. These conditions, of course, still obtain in Britain, but the more exacting overseas and military requirements assume increasing importance.

It should not be presumed, however, that exhaust braking is necessary or desirable only for vehicles operated continually over long gradients. It can effect substantial operating economies by reducing wear on the wheel brake shoes and drums and also lowering the time and cost of maintenance and adjustment. This is well shown in Switzerland where exhaust braking is employed not only by the Postal Authority for their vehicles operating over the mountain passes but also by municipalities for urban passenger transport systems. It is claimed that shoe and drum wear can be reduced by from 25 to 50 per cent according to operating conditions.

Other advantages accrue. Driver fatigue is reduced as fewer gear changes and fewer and less sustained applications of the wheel brakes are required. It can be used for checking speed when in dense traffic or when cornering, and on greasy or icy road surfaces the smooth application of braking effort through the transmission minimizes the risk of wheel skids.

Exhaust braking also contributes substantially to the safety of a vehicle. It can form a completely independent self-contained system additional to the conventional wheel brakes and is not susceptible to fading. Down grades can be descended at a steady speed on the exhaust brake alone without accelerating the vehicle or overspeeding the engine and, should an emergency arise, the wheel brakes are cool and in the most effective condition. Incidental operating economies are effected as fuel is cut off during exhaust brake application, the working life of tyres is extended by reason of smooth braking and the virtual impossibility of locking the wheels, and engine and transmission wear is reduced since rotational speeds are lower during descents.

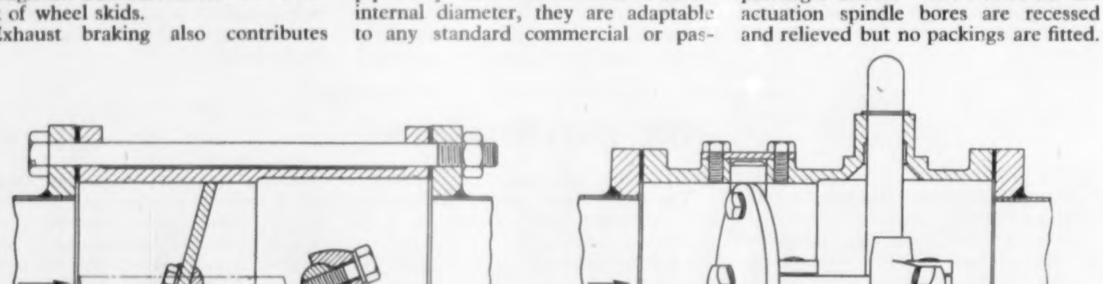
The control valve for the independent hand-operated system is mounted on the steering column

The well-proved Oetiker brake is now produced in this country by Parkwood (Brakes) Ltd., Sunbury-on-Thames, Middlesex, who have acquired from the Swiss parent company the sole manufacturing rights for the British Empire. Made in two sizes, for exhaust pipes up to 2·125 in and 3·125 in internal diameter, they are adaptable to any standard commercial or passenger vehicle. The location is not critical and, provided a suitable control is arranged, the unit may be sited at any point in the pipe running from the exhaust manifold to the silencer.

In construction it is simple and robust. It consists of a malleable iron cylinder, flanged at each end for attachment by three through-bolts to complementary flanges welded to the ends of the divided exhaust pipe. Flat asbestos washers are provided at the joint faces. Two transverse holes are bored in the cylinder, one for the spindle of the valve and the other, located downstream of the valve, for the actuation spindle. The steel spindle carrying the malleable iron butterfly valve takes its bearing in the unbushed bored holes with generous clearance both diametrically and laterally so that the valve is permitted a limited float to seat positively when it closes at an angle of 15 deg to a plane normal to the axis of the cylinder. Steel blanking plates, each attached by two set screws, close the bored holes.

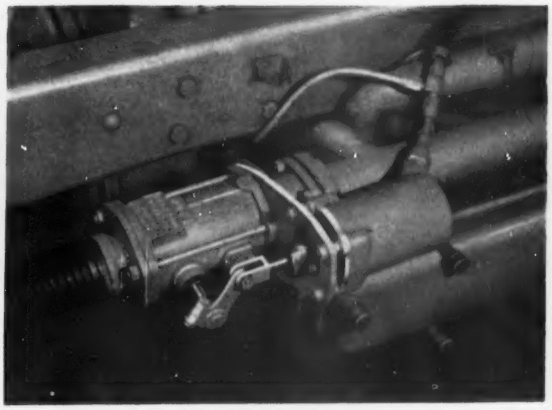
The actuation spindle, laterally located by spring clips, projects through its unbushed borings to receive at either side the operating lever. Forked members bolted to both the valve and the actuation spindle are connected by a steel link. A radial movement of 75 deg by the actuation spindle brings a stop, formed integrally with one arm of the actuation spindle fork, into contact with the butterfly valve and prevents movement of the valve beyond a plane containing the cylinder axis.

This method of mounting and actuation is a patented feature, ensuring that noxious gases cannot escape by way of the valve spindle and possibly find their way to the driver's cabin or the passenger saloon. The bosses for the actuation spindle bores are recessed and relieved but no packings are fitted.



Arrangement of the Oetiker exhaust brake. The relay method of actuating the butterfly valve obviates gas leakage

Exhaust brake, with air pressure operating cylinder, located in exhaust pipe below propeller shaft



Convenient mounting of exhaust brake on underfloor engine of Leyland Cub chassis

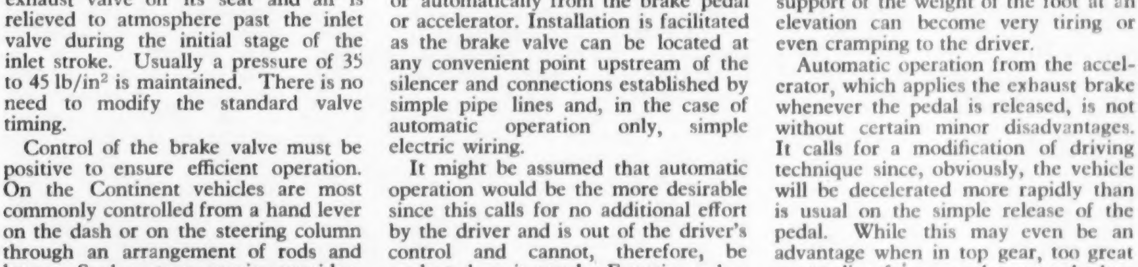
They are not necessary, in fact, since any gases forced past the closed valve will drop to atmospheric pressure and pass out through the silencer rather than escape past the spindle. Exhaust manifold and pipe joints up to the brake valve must, of course, be capable of holding an internal pressure of up to about 50 lb/in^2. The actual pressure established will depend upon the characteristics of the engine and is determined by the spring loading of the exhaust valve and the overlap period. Pressure will rapidly be built up in the manifold and pipe until it reaches a value sufficient to hold the exhaust valve off its seat and air is relieved to atmosphere past the inlet valve during the initial stage of the inlet stroke. Usually a pressure of 35 to 45 lb/in^2 is maintained. There is no need to modify the standard valve timing.

Control of the brake valve must be positive to ensure efficient operation. On the Continent vehicles are most commonly controlled from a hand lever on the dash or on the steering column through an arrangement of rods and levers. Such systems require considerable effort to operate and are liable to need fairly frequent adjustment. Furthermore, should the exhaust manifold of a forward-mounted engine be on the opposite side of the vehicle to the driver, or should the engine be located under the floor, the linkage tends to become unduly complicated.

Since most commercial vehicles are equipped with either vacuum or air pressure braking systems it would seem logical to utilize the available power to operate the exhaust brake. The Clayton Dewandre Co. Ltd., of Lincoln, in co-operation with the makers of the Oetiker brake, have designed and developed a comprehensive range of power-operated control equipment. For both vacuum and air pressure systems control can be arranged independently by hand lever or automatically from the brake pedal or accelerator. Installation is facilitated as the brake valve can be located at any convenient point upstream of the silencer and connections established by simple pipe lines and, in the case of automatic operation only, simple electric wiring.

It might be assumed that automatic operation would be the more desirable since this calls for no additional effort by the driver and is out of the driver's control and cannot, therefore, be neglected or ignored. Experience has shown, both here and on the Continent, that an independent hand control is preferred by the majority of drivers. Where automaticity is arranged from the normal brake pedal the exhaust brake is operated as a result of a slight movement, requiring only a light pressure, of a special pad hinged over the pedal. This may be difficult to assess precisely, particularly if the foot has to be raised some distance from the footboard to reach the pedal, as is not uncommon. Consequently, the movement may be excessive and a light pressure may inadvertently be applied to the friction brakes also. Should this happen regularly some unnecessary wear of the linings will occur, and in each instance the advantage of cool friction brakes will be lost. Furthermore, on long gradients, the continued support of the weight of the foot at an elevation can become very tiring or even cramping to the driver.

Automatic operation from the accelerator, which applies the exhaust brake whenever the pedal is released, is not without certain minor disadvantages. It calls for a modification of driving technique since, obviously, the vehicle will be decelerated more rapidly than is usual on the simple release of the pedal. While this may even be an advantage when in top gear, too great a retarding force may be exerted when in a lower gear in traffic or on a modest gradient. Unless a change be made to a higher gear, some operation of the accelerator pedal, and consequently of the exhaust brake valve, may be neces-



Assembly of exhaust brake and air pressure cylinder (full line) or vacuum cylinder (broken line)

sary to keep the vehicle rolling. Under such conditions, should an emergency stop be called for, simultaneous withdrawal of the clutch and application of the friction brakes would inevitably result in "losing" the engine. A similar situation could arise, of course, with automatic operation from the friction brake pedal or even with hand operation. With the last method, however, such a contingency could be avoided by an immediate opening of the exhaust brake valve.

It is possible to adjust the butterfly valve and the fuel cut-off so that the engine will idle when the clutch is withdrawn but opinion on the advisability of this method is somewhat conflicting. The makers of the exhaust brake recommend, and a number of engine manufacturers insist, that fuel injection be cut-off with application of the brake. To some extent diversity of opinion arises from varied appreciation of the function of the exhaust brake. It was designed primarily as a retarder and not as a stopping brake.

For the independent hand control system a valve is mounted on the steering column, clear of the driver's knee, for either right-hand or left-hand operation. Air pressure from the reservoir is piped to the valve and from the valve a lead is taken directly to the operating cylinder mounted on the exhaust brake valve. A branch from this lead is taken to the fuel injection cut-off. The installation for a vacuum system is similar but with appropriate change of type of components and re-positioning of the external lever on the exhaust brake valve for a pull stroke instead of a push stroke. The simultaneous cut-off of fuel is arranged by means of a small air power cylinder or a small diaphragm chamber for pressure or vacuum systems respectively. Each of these devices has a stroke of 0·75 in and a terminal pull of about 20 lb.

In automatic systems control is effected by a solenoid-operated air pressure or vacuum valve. The solenoid is energized from the vehicle battery when a Micro switch is operated from the brake pedal or from the accelerator. As only a small solenoid is necessary to actuate the control valve, the electric current is very low and in no way comparable to the consumption of the large solenoid used to close the exhaust brake valve directly in an all-electric system.

Test runs on a Leyland 7·4 litre engined vehicle weighing about 9 tons demonstrated the efficacy of the Oetiker exhaust brake as a retarder on an urban bus route with considerable traffic in both directions. The hill, about half a mile long with a bend at about one-third distance, has an average gradient of 1 in 9. The surface was good and in dry condition and for each run the approach was at a speed of about 25 m.p.h. First a normal run in third gear was made using only the friction brakes. Continuously a firm application of the brakes was necessary to hold the vehicle under control at a speed just below 20 m.p.h.

During an opportune lull in the traffic flow a descent was made in third gear with the vehicle running freely without any brake application. The engine was raced above its governed speed and the vehicle accelerated to 30 m.p.h. With minor fluctuations, above and below, that speed obtained throughout the descent. Again in third gear, the vehicle made a descent using only the exhaust brake. Speed dropped almost immediately to 20 m.p.h. and was steadily maintained at slightly below that speed. The efficiency of the exhaust brake when a lower gear ratio is used was demonstrated in a run in second gear. Speed was brought down to less than 5 m.p.h. and three times during the descent the clutch had to be withdrawn to keep the vehicle rolling.

The tabulated performance figures were taken from a series of tests conducted by a prominent French vehicle manufacturer on a 10-ton lorry having a six-cylinder engine of 120 mm bore and 160 mm stroke and fitted with the Oetiker exhaust brake. The gross weight of the vehicle was 41,900 lb.

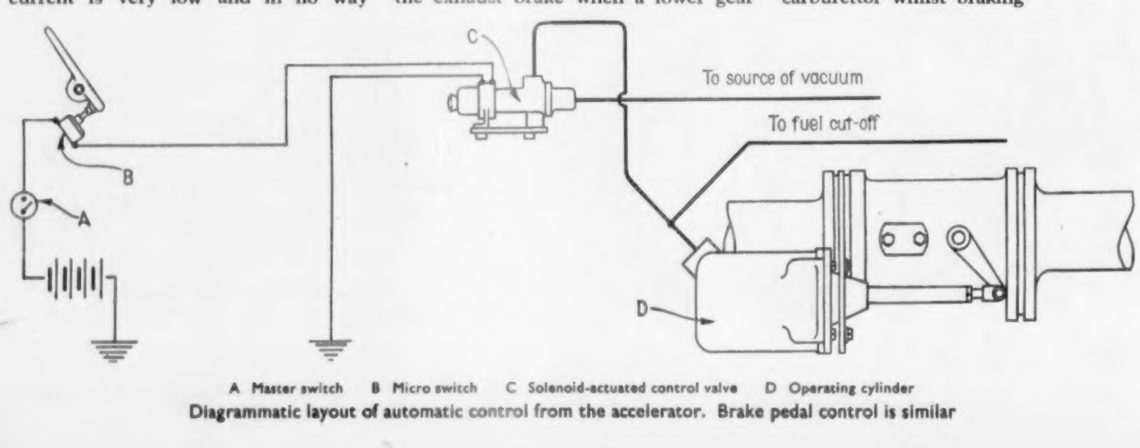
Reference throughout has been to diesel-engined vehicles, for which the brake is specifically intended but an exhaust brake can be fitted to a petrol engine. The performance may be slightly lower since the petrol engine, operating on a lower compression ratio, necessarily has a larger clearance volume and is thus less effective when used as a compressor. This will not, however, nullify its usefulness. Installation is more complicated as it is not possible to fit a simple fuel cut-off device on an orthodox engine aspirating a carburetted fuel-air mixture. Accordingly, an auxiliary air intake pipe, complete with an air cleaner, is branched off the intake downstream of the carburettor and a change-over valve, actuated simultaneously with the brake valve, is arranged to by-pass the carburettor whilst braking

TABLE I. LORRY, 18¾ TONS GROSS, ON LEVEL

Gear	Speed at time of braking (m.p.h)	Stopping distance (yards)	Deceleration rate (ft/sec²)
5th	31·0	437	0·78
4th	26·1	273	0·89
3rd	15·5	55	1·57
2nd	6·8	10	1·67

TABLE II. LORRY, 18¾ TONS GROSS, ON DOWN GRADES

Gradient (per cent)	Gear	Speed at time of braking (m.p.h)	Limit speed of deceleration	Deceleration rate (ft/sec²)
4·5	4th	18·6	18·6	0
8	4th	21·7	18·6	0·052
9	3rd	15·5	6·2	0·109
10·5	3rd	15·5	15·5	0



A Master switch B Micro switch C Solenoid-actuated control valve D Operating cylinder

Diagrammatic layout of automatic control from the accelerator. Brake pedal control is similar

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

A "STAFFA" $\frac{3}{8}$ in single spindle screwing machine designed and built by Chamberlain Industries Limited, Staffa Road, Leyton, London, E.10, is illustrated in Fig. 1. The pedestal of the machine is of fabricated construction and comprises two side panels pressed-out from mild steel sheets that form an integral part of the cutting lubricant and swarf cradle. This obviates joints in the pedestal construction through which lubricant could escape.

The machine has a hollow spindle upon which a rotating diehead can be fitted. It runs in self-lubricating Oilite bearings with thrust washers of the same material to take end loads during the cutting operation. A small gear pump, housed within the reservoir, pumps the cutting fluid. Any leaks from the pump are trapped in the reservoir and cannot contaminate the belts and driving pulleys. The cutting lubricant is delivered to the work through the hollow spindle in such a manner that the diehead is washed clean of swarf and both sides of the chasers are kept clean to ensure maximum cutting efficiency. Both the spindle and the lubricant flow-control fittings are totally enclosed at the top of the machine by a removable push-on cover. To ensure cleanliness and complete protection for the operator against lubricant and revolving components, the diehead itself is enclosed by a hinged cover.

A swarf trough with a fine gauge wire mesh filter is mounted directly under the mouth of the cutting lubricant and swarf cradle. For further filtration of the lubricant before it is pumped back to the work, there is a cylindrical wire mesh filter in the suction pipe of the pump.

The carriage assembly incorporates a flat top table to which any form of self-centring vice or work-holding fixture can be mounted. Two tubular guide bars, running the length of the cradle, provide the slide bed. They are extended on either side of the front bearing housing. These guide bars maintain the table top in correct vertical and horizontal alignment with the centre of rotation of the machine spindle. Coupled to a simple linkage at the rear end of the cradle, the forward and return movement of the carriage assembly is controlled by the action of a lever that swings through 90 deg. There are no cams, gears or racks and pinions.

Two screwed shafts with trip adjusting nuts are housed in the tubular slide bars to allow the length of thread to be cut to be pre-set. A crosshead bracket positioned inside the machine pedestal couples the screwed shafts together, and from this crosshead bracket the yoke of the diehead is automatically controlled.

Drive to the spindle is by single vee belt. The variable drive assembly is mounted on a quadrant shape bearing, and speed variations to suit different materials are obtained by moving the quadrant up or down. Quadrant operation is effected by means of a control handle at the side of the machine. A pointer indicates on a graduated scale the speed at which the diehead is running. The speed range is 266 to 1,007 r.p.m.

Fig. 1. Staffa $\frac{3}{8}$ in single spindle screwing machine
Chamberlain Industries Ltd.

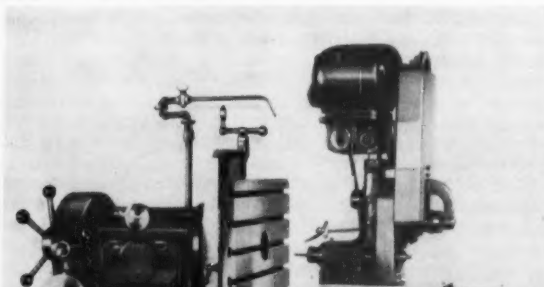


Fig. 2. Beichle universal toolroom milling machine
The Elgar Machine Tool Co. Ltd.

Universal milling machine

The Beichle universal tool room milling machine illustrated in Fig. 2 incorporates design features that make it suitable for producing a very wide variety of profiles and also allow work to be machined on several sides at one setting. The vertical table is mounted on a slide on the horizontal main spindle. Both the table and the slide can be turned through 360 deg independently of each other. It is therefore possible to offset the table up to a maximum of $5\frac{7}{16}$ in. Thus, by rotation of the main spindle, radii from 0-5 in and straight surfaces can be milled in one continuous operation.

The standard equipment includes a 24 notch direct indexing plate on the main spindle and three indexing plates for mounting on the worm spindle. With these practically any number of divisions may be obtained. In addition, an indexing head of 4 in centre height and $13\frac{3}{4}$ in between centres can be mounted on the table. This equipment makes it possible to mill conical or taper components with either plain or serrated surfaces. The table has tee slots and a central bore and can accommodate a three-jaw chuck with adaptor plate, centring spigot and index pin, or a self-centring parallel jaw vice with reversible jaws.

A hollow, tubular section main spindle is employed. It will accept bars

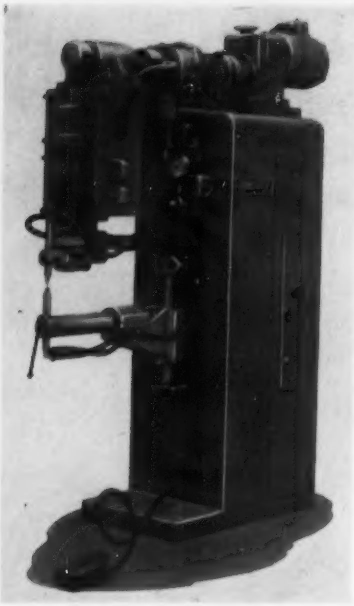


Fig. 3. Motor-driven pitch-spot and seam welding machine
Sciaky Electric Welding Machines Ltd.

up to $2\frac{3}{4}$ in diameter which may be held in the three-jaw chuck. The hollow spindle is also provided with a No. 8 metric taper socket and a sleeve that will take collets up to $\frac{3}{4}$ in diameter. The cutter spindle headstock is mounted on a cross slide that can be swivelled 30 deg to either side. Mounted on the cross slide is the top slide, which can be swivelled through 90 deg to either side and which carries the headstock. This, in turn, can be swivelled through 180 deg. Long thin work, such as shafts and punches, can be machined both radially and longitudinally by setting the headstock over at an angle of 90 deg. A $1\frac{1}{2}$ h.p. reversible motor drives the cutter spindle through a four-step vee belt pulley to give a speed range of 95-2,000 r.p.m.

This machine is available in three sizes. Its greatest advantage is that work can be machined at any angle and that radii that blend with straight surfaces can be milled in one continuous cut with standard end mills. The wide range of adjustments to both the table and the spindle stock make the machine extremely versatile. The sole British agents for these machines are The Elgar Machine Tool Co. Ltd., Hampton Road West, Hanworth, Feltham, Middlesex.

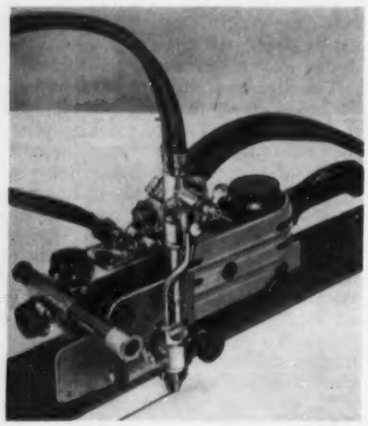


Fig. 4. Flame cutting machine
Oxhycarbon Co. Ltd.

Stitch-seam welder

A recent addition to the range of welding equipment manufactured by Sciaky Electric Welding Machines Ltd., Farnham Road, Slough, Bucks, is illustrated in Fig. 3. It is the Sciaky S.M.C.100 motor-driven pitch-spot and seam-welding machine. Features incorporated in the machine make this machine suitable for welding the heat-resistant steels. As can be seen from the illustration, ram motion feed is applied to the top electrode by means of a special assembly contained in the head block. This assembly comprises a pneumatic cylinder with a short connecting rod coupled to a slide block that holds the top electrode assembly.

Drive to the eccentric coupling is by a $2\frac{1}{2}$ h.p. motor through an infinitely variable speed gear that embodies a brake-and-clutch mechanism. From this stage, the transmission passes through a fixed-reduction gearbox to the eccentric shaft on which is mounted a pair of cams for controlling limit switches, and thence finally to the eccentric wheel for electrode pressure-thrust. A tumbler switch is used for running the motor up to operating speed. For operating the machine a two-position foot-switch is used.

When the top electrode is in the fully retracted position, the gap between the electrodes must be $1\frac{3}{4}$ in plus an amount equal to the combined thickness of the material to be welded. The amount of travel in excess of $1\frac{3}{4}$ in has to be maintained at the minimum value, since it represents the welding stroke.

An interesting feature of this machine is a compensating air-circuit that includes a reservoir fitted in series with the head cylinder and air-pressure control. As the reservoir has a capacity in the order of 730 in^3, while the eccentric operates through only $1\frac{1}{4}$ in of thrust—representing about 29 in^3 of the capacity in the head cylinder—the air pressure in the cylinder due to the downward thrust of the cylinder towards the stationary piston dissipates into the reservoir, thereby keeping the first half of the pressure-cycle at a constant level. On the upward stroke of the eccentric, the excess pressure in the reservoir returns to the head cylinder to keep the second half of the cycle constant. The welding current initiation cam is adjustable to allow the weld-time period to be set to the most suitable position within the pressure/time curve.

When the machine is set for single-spot welding and the top electrode is closed down on top of the work piece, a pressure switch operated by the pressure build-up prevents any rotation of the eccentric until the top electrode is down to its full welding position. As soon as the top electrode reaches this position, the foot-switch is depressed to its second position where it releases the eccentric wheel-brake and engages the clutch to allow the eccentric to rotate. When the eccentric has rotated through a few degrees, a control switch is closed by a cam on the eccentric shaft and the pressure is applied to the weld area. After a further few degrees of rotation, a second cam closes to initiate the welding current. A synchronous Ignitron panel controls the heat and duration of the welding current.

For stitch-seam and spot welding, the change-over switch is set to the auto-continuous position. The operation sequence is similar to that for the individual spot welding cycle, with the exception that the foot switch is kept depressed.

Automatic continuous welding can be carried out, with infinitely variable control for 20 to 120 welds per minute. It should be noted that the maximum speed of welding is dependent upon eccentric drive-gearbox ratio, and if desired the machine can be arranged for a maximum of 300 welds per minute. By a series of tappings with vernier control between ranges, the welding current is infinitely variable up to a maximum of 23,500 amps. To provide consistency in welding current despite any fluctuation in line voltage, the Ignitron control panel has a constant current regulator.

Flame cutting machine

Oxhycarbon Company Limited, 6, Hainthorpe Road, West Norwood, London, S.E.27, have recently added the "Quicky B" machine shown in Fig. 4 to their range of small, inexpensive flame cutting machines. It is an improved version of the original "Quicky" flame cutter. Among the improvements that are embodied in the design are a more powerful electric motor, an improved gearbox with worm wheel drive, and means for raising or lowering the cutter whilst the machine is cutting.

It weighs only 13 lb and is claimed to be the smallest flame cutting machine on the market. Despite its small size and light weight it can be used for cutting steel from $\frac{1}{16}$ in to $2\frac{1}{2}$ in thick. Normally power is supplied by a 14 watt, 230 volt A.C/D.C. motor, but if desired a 110 volt A.C/D.C. machine can be supplied. Speed control is by a simple rheostat which is graduated directly in plate thickness to be cut.

Normally the "Quicky B" is supplied complete with a circle cutting trammel for cutting circles from $2\frac{1}{2}$ in to 40 in diameter, a 3 ft guide rail for straight line cutting and a cutter with a box of nozzles. Provided the correct nozzles are fitted, the cutter will work with acetylene, propane, coal gas or hydrogen.

MIXTURE SAMPLING

Development of a Method for Fuel and T.E.L. Distribution Study

A. L. Wachal, Dipl.Ing., A.M.I.Mech.E., M.Inst.Pet.

IN petrol engines the intake system serves the purpose of preparing a combustible air and fuel mixture and of distributing it as evenly as possible among the cylinders. Unless resort is made to a multi-carburettor system the difficult task of distribution is seldom accomplished in a completely satisfactory manner, with the result that both qualitative and quantitative mal-distribution of fuel and air are common in multi-cylinder automotive engines. It is the light components of the fuel and the heavy materials including the lead base anti-knock additives that are primarily subjected to uneven relative distribution. In consequence, the fuel entering individual cylinders can vary appreciably in anti-knock quality, and to prevent detonation the fuel octane number has to be raised to the requirement of the cylinder which receives the most knock-prone air/fuel mixture.

Mixture formation in the induction system is not a simple problem; factors depending upon both the design and running conditions, such as the aerodynamic and temperature conditions in the carburettor and intake manifold, are major considerations; but the properties of the fuel, especially its volatility, also play prominent parts in determining both the quantities of fuel and air and quality of mixture reaching each cylinder. Investigations into these factors are usually hampered by difficulties encountered in sampling the mixture from each cylinder and in analysis of the air/fuel sample. There have been many attempts to overcome these difficulties but none of the methods seems to cope successfully with all aspects of the problem. The unburnt air/fuel mixture for distribution study can be sampled either before or after reaching the cylinders of an engine.

In the first alternative the combustible mixture is drawn from the induction system at each cylinder inlet port[1]. Samples thus obtained, however, are seldom truly representative as the least volatile fraction of fuel which has not completely evaporated tends to flow along the manifold walls. By sampling directly from each cylinder, this disadvantage can be avoided, but this method calls for special cam or electrically operated sampling valves[2,3] and can hardly be envisaged without considerable modification to the engine. The more favoured methods for the assessment of mixture strength and lead distribution depend upon exhaust gas sampling from the engine exhaust ports[4,5,6]. These methods are also not entirely satisfactory as exhaust gas sampled from the conventional exhaust manifold is invariably contaminated by products of combustion from other cylinders, even when the sampling point is as near as practicable to the exhaust valve. The use of an individual exhaust pipe for every cylinder cannot be arranged without serious departure from normal conditions both at the hot spot and in the exhaust system as a whole and changes to these will affect carburation. In all the methods discussed even more serious difficulties are encountered in the actual analysis of the gaseous samples thus obtained.

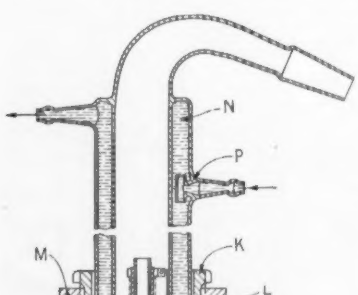


Fig. 2. Sampling plug assembly

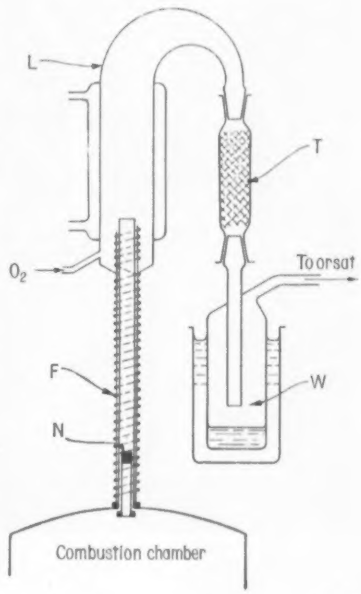


Fig. 1. Diagrammatic arrangement of lead sampling apparatus. N—sampling nozzle. F—tubular furnace. O_2—oxygen feed inlet. L—water cooled lead precipitator. T—glass wool packed lead trap. W—exhaust water condenser

Apart from mass spectrometry there are no suitable means for direct analysis of the unburnt air/fuel mixture and even this method is not altogether satisfactory when high molecular weight hydrocarbons and T.E.L. are in question. In such a case mixture quality can be determined only by analysis of the fuel quantitatively separated from the accompanying air. This latter operation is by no means a simple one and can be carried out efficiently only by employing a rather involved trapping system. Furthermore, reasonable accuracy in the determination of the amount and the nature of fuel extracted from the sampled gas can only be expected when the quantity of fuel trapped is quite appreciable. This, in turn, means long sampling periods which limits the applicability of the method to cases in which these conditions can be met. It is also difficult to adapt this method for lead distribution investigations.

Petrol engine exhaust gas is also not easy to deal with analytically, mainly because combustion in petrol engines is never quite complete and appreciable quantities of decomposed and partially burnt fuel are found in the outgoing gases[7]. This makes the Orsat apparatus of very limited value since it fails to identify and measure the quantity of the complex constituents in petrol engine combustion products. In this respect complete oxidation prior to analysis of petrol engine exhaust gas offers considerable help. For this purpose various oxidizing agents have been used[8,9], but an oxygen aided combustion has been confined to analytical laboratories as an essential step in the gravimetric analysis of unburnt or incompletely oxidized air/fuel mixture.

A new sampling method

In the sampling method to be described, which was designed for the study of T.E.L. and mixture distribution in a multi-cylinder engine, furnace oxidation forms an

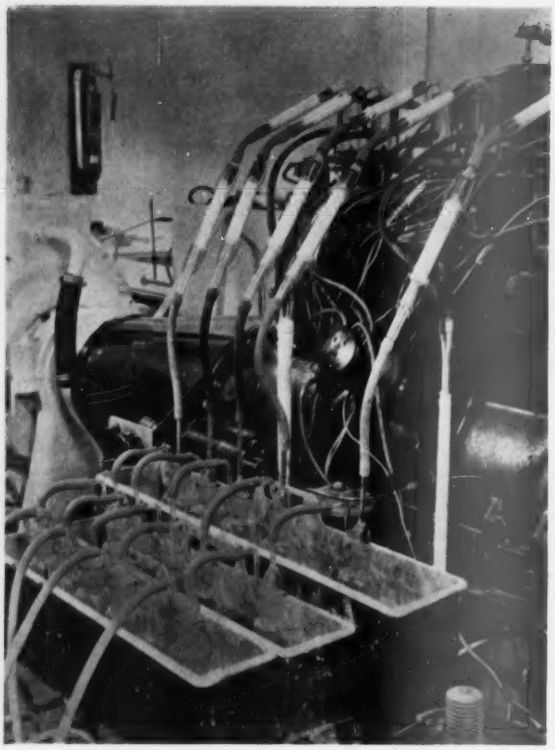


Fig. 3. Sampling apparatus set up on six-cylinder engine

integral part of the apparatus. The main object in developing a new sampling method was to overcome difficulties inherent in the techniques already mentioned. The sampling apparatus is shown diagrammatically in Fig. 1. It embodies the following principles:

(1) A sample of the cylinder charge is taken continuously from each combustion chamber through an orifice N.

(2) To prevent uncontrolled deposition of lead compounds before the gas sample reaches the trapping system, the sampling orifice N is placed in a heated zone of a tubular passage F.

(3) The electrically heated miniature furnace F serves the purpose of igniting the unburnt portion of gaseous mixture which passes through the sampling orifice.

(4) A sufficient quantity of oxygen or air is fed to the sampling apparatus to enable complete oxidation of ignited combustible.

(5) A cooled and suitably shaped glass surface L is provided for precipitation of lead compounds from the hot gases leaving the tube F.

(6) The water in the outgoing gases due to atmospheric humidity and from combustion, is trapped in a series of condensers W; the quantity can be accurately measured.

(7) From the amount of lead collected in the trap L and from the quantity of condensed water the lead content of the fuel from each cylinder can be calculated.

(8) The exhaust gas, which leaves the system dry and free from complex combustion products, can be analysed in a simple Orsat apparatus for determination of the air/fuel ratios in each cylinder.

TABLE I. PERCENTAGE OF LEAD RECOVERY FROM GAS DRAWN CONTINUOUSLY FROM THE COMBUSTION CHAMBER OF A C.F.R. ENGINE

Sampling orifice diameter 0·0136 inch.
Reversed stream type lead collector without glass wool taps.

Fuel Used	Recovered Quantities of		Percentage of Lead Recovered	Deviation from Average	Remarks
	Lead %	Water g	%	%	
Technical iso-octane +3 ml. T.E.L./I.G.	0·00595	16·8	49·2	—3·4	Water/fuel ratio from exhaust gas analysis 1·38 (theor. 1·42)
	0·0127	33·0	51·1	—1·5	
	0·0111	30·0	49·2	—3·4	
	0·0147	34·2	56·8	+4·2	
+1·5 ml. T.E.L./I.G.	0·0151	73·2	54·7	+2·1	Average lead recovery 52·6 per cent
+4 ml. T.E.L./I.G.	0·025	47·7	54·5	+1·9	

In the sampling unit developed on these lines and shown in Fig. 2, the heater B was manufactured from stainless steel tube of $\frac{3}{16}$ in O.D. by $\frac{1}{8}$ in I.D. and about 4$\frac{5}{8}$ in long. The inside diameter of the tube was enlarged in two steps, this being found advantageous for stabilizing the flame at the outlet of the tube. The heater winding consisted of approximately 10 ft of 0·031 in by 0·006 in Brightray strip. It was fed from the Variac auto-transformer and at the heater operating temperature of about 800 deg C consumed about 360 watts. The insulation between the winding and the tube was originally asbestos paper base and two layers of mica strip. Heaters constructed in this way had, however, rather short and unpredictable life. They usually become unserviceable owing to local failure of the insulation followed by fusion of the tube and winding at the hottest part of the tube. The insulation was then replaced by a silica sleeve which did not show these faults. Lavish initial clearance between the silica sleeve and the heater tube was necessary to compensate for the low coefficient of expansion of silica and distortion of the heater tube during use.

To provide satisfactory combustion conditions the dimensions of the heater, its operating temperature and the sampling rate, have to be closely inter-related. It must be appreciated that, for a given heater, there is only a limited range of sampling rates at which ignition of the gaseous stream can take place and the flame remains stable at the outlet from the heater. An undue increase of the flow rate results in the blowing out of the flame; a decrease causes

Fig. 4. Sampling apparatus for single-cylinder engine

TABLE II. PERCENTAGE OF LEAD RECOVERY FROM GAS DRAWN CONTINUOUSLY FROM THE COMBUSTION CHAMBER OF A C.F.R. ENGINE

Sampling orifice diameter 0·015 inch.

Fuel Used	Recovered Quantities of Lead g*	Recovered Quantities of Exh. Water g.	Percentage of Lead Recovery %	Deviation from Average %	Remarks
Technical iso-octane +3 ml. T.E.L. Per I.G.	0·0349	40·6	63·0	—3·3	Water fuel ratio 0·977
	0·0723	84·8	62·7	—3·6	
	0·0909	88·0	75·9	+9·6	
	0·0667	78·6	62·4	—3·9	Average lead recovery 66·3 per cent
	0·0538	58·9	67·3	+1·0	

*Lead/fuel ratio equals 0·000347 per ml. T.E.L. per I.G.

it to settle inside the heater tube, which then burns out in a very short time. For the heater dimensions given above, the permissible sampling rate was found to range from approximately 1·5 to 2·5 litres per minute. This requires a sampling orifice not exceeding 0·013 in. in diameter with a length of 1/16 in. Some difficulties were experienced in manufacturing this size of orifice in stainless steel, particularly as it was essential that the bore was concentric with the outside. These difficulties were overcome by the use of stainless steel capillary tube of the correct dimensions, thus avoiding the need for drilling small diameter holes.

The Pyrex glass lead precipitator N used with the sampling unit was water cooled. The water inlet pipe P was directed axially so as to prevent local overcooling and condensation of the exhaust water on the surface where lead compounds were precipitating. For the same reason the inlet water temperature was kept very close to boiling point.

To keep the speed of gas passing the trapping system as low as possible, an oxygen feed was used in preference to air. The feeding plug with a suitable size orifice E was located in the upper part of the sampler body opposite the heater terminal D. Oxygen was supplied under constant pressure from a high pressure cylinder through a two-stage reducing valve.

Some manufacturing details of the apparatus can also be seen in Fig. 2. The sampling nozzle A has an external thread on to which the heater is screwed making a seal against the sampler body through a copper gasket G. The body C is machined from stainless steel to allow acid to be used to remove lead compounds if any deposition occurred on the cool surface. The common strong-back M was used in conjunction with the ring unit K and the copper gasket H to provide the gas-tight seal between the machined face in the cylinder head and the base of the sampler body. The external water seal is made by tightening the nut F against the mild steel and rubber rings J fitted to the countersunk hole in the upper wall of the cylinder head. The lead precipitator N was attached to the sampling unit by a clamp with two adjusting screws which can be seen in Fig. 3; an asbestos paper gasket (L, Fig. 2) being used to obtain a gas seal between the base of the precipitator and the top of the sampling unit. Fig. 3 shows the general view of the sampling apparatus on a 6-cylinder engine; two glass wool packed tubes attached to the precipitator were used as secondary lead traps. These were connected to the water trapping system which consisted of three standard traps cooled in ice, ice and salt and solid CO_2 respectively. The outlets from the water trapping system were connected to displacement type gas meters which measured the total volume of gas sampled from each cylinder.

TABLE III

ENGINE DATA	
Type	Side Valve
Number of Cylinders	6
Bore	85 mm.
Stroke	120 mm.
Total Cylinder Capacity	4085·7 cc.
RAC Rating	26·88 h.p.
Brake horse power	75/80 at 3000 r.p.m.
Compression ratio	6·07
Cylinder head	Cast Iron
Piston	Aluminium Alloy
Carburettor	Zenith Stromberg down draught
SPECIMEN ENGINE CONDITIONS	
Speed	1000 r.p.m.
Water outlet temperature	162°F.
Oil sump temperature	190°F.
Throttle	full

TABLE IV. PERCENTAGE OF LEAD RECOVERY AND AIR/FUEL RATIOS OBTAINED ON 6-CYLINDER PETROL ENGINE

Sampling orifice diameter 0·0130 inch.

Fuel Used	Cylinder No. 1	2	3	4	5	6	Mean from all Cylinders	
Technical iso-octane +3ml. T.E.L. Per I.G.	–	Not Sampled	–	–	–	71·7		Initial single sampler test
	52·5		64·3	63·4	52·6	66·5	59·9	
	61·2		54·8	65·9	53·6	70·8	61·3	
	48·0		51·8	67·9	40·9	75·8	56·9	
Mean from all Tests	53·9		57·0	65·7	49·0	71·0	59·3	
Air-Fuel Ratio*	15·1	–	13·2	13·3	13·6	14·9	13·9†	

*Figures quoted are mean values from three tests.

†For the calculation of the mean air/fuel ratio for the whole engine, the air/fuel ratio in cylinder No. 2 was taken equal to that from cylinder No. 5.

Development of the method

Materials are used in conjunction with lead tetra-ethyl with the object of evacuating the lead compounds after combustion. It was, therefore, of primary importance to know what percentage of lead originally present in the fuel could be recovered from the engine combustion chamber by means of the sampling apparatus employed and the repeatability of the results thus obtained.

TABLE V. ANALYSES OF OUTLET GAS WITH AND WITHOUT OXYGEN FEED

Percentage by Volume	Without Oxygen Feeding			With Oxygen Feeding		
Cylinder No.*	3	4	5	3	4	5
Carbon Dioxide	11·87	7·44†	12·31	14·14	14·97	12·62
Carbon Monoxide	4·80	5·24	3·41	0·84	0·65	0·57
Oxygen	Nil	5·72	0·09	9·20	8·8	17·65
Methane	0·04	1·70	0·09	Nil	0·04	0·06
Hydrogen	1·75	2·16	1·39	Nil	Nil	Nil
Formic Acid	0·02	0·02	0·01	Nil	Nil	Nil
Formaldehyde	0·01	0·03	0·06	Nil	Nil	Nil
Unburnt Fuel	0·23	0·01	0·01	Nil	Nil	Nil

*Only rich mixture cylinders included.

†Example of outlet gas composition when heater failed to ignite the unburnt portion of the sampled gas.

The initial work on these lines was carried out on the single cylinder C.F.R. engine in conjunction with the sampling apparatus shown in Fig. 4. No oxygen feed was provided to aid the combustion and a reversed stream type of lead precipitator was employed in this sampling unit. Tables 1 and 2 show the results obtained with leaded technical iso-octane as fuel. Results presented in Table 2 were obtained with an increased sampling rate and these show a somewhat higher percentage of lead recovery; the low value of the water/fuel ratio (0·977) as obtained from the complete analysis of exhaust products indicates that under these conditions only a partial combustion was taking place.

Further lead recovery tests were carried out on a 6-cylinder standard production type engine. Data for this engine and specimen running conditions are given in Table 3. In Fig. 3 it will be noticed that only five sampling units are fitted. This was due to difficulties in fitting a sampling unit to No. 2 cylinder where the water outlet was located. As earlier work had shown that air/fuel ratio distribution was almost symmetrical about the centre of the engine it was thought that the results obtained from No. 5 cylinder would sufficiently well indicate the behaviour of cylinder No. 2.

Lead recovery figures, together with the mean results obtained from three tests carried out on this engine are presented in Table 4.

To check the method for efficiency of lead trapping all parts of the sampling equipment on which any deposition of lead compounds could occur were examined quantitatively for lead. The data thus obtained showed that the lead recovered was distributed throughout the trapping train as follows:

Water cooled precipitator	64·5 per cent
Glass wool filters	35·1 per cent
Exhaust water traps	0·4 per cent
Inner surface of sampling unit body ..	nil
Sampling orifice and heater tube ..	nil

It will be appreciated that the efficiency of combustion in the tubular heater is of basic importance in the use of this method. If combustion is incomplete then unburnt or partly oxidized constituents will be present in the gas leaving the apparatus. Under such conditions it would be necessary to employ during each sampling period a complicated and not wholly reliable chemical analysis for the determination of combustible constituents, and to establish from this a true water/fuel ratio for the combustion which had produced the water trapped during the test. If the combustion is complete the theoretical water/fuel ratio for the fuel burnt can be used, as all the hydrogen will be combined with oxygen to form water.

To establish sampling conditions conducive to high oxidation efficiency a series of experiments was carried out first on a single cylinder C.F.R. engine and later on the 6-cylinder engine. Table 5 gives the analytical results for combustion efficiency tests with and without oxygen feed obtained from the richest cylinders of the 6-cylinder engine. It is clear that the oxygen feed is essential and that under oxygen-aided sampling conditions the high combustion efficiency is manifested by the absence of partially oxidized hydrogen-bearing constituents and by very low carbon monoxide content in the gas samples. The water collected during the test, after correction for atmospheric humidity, was used to establish the quantity of fuel burnt to give the lead deposits found in the trapping train.

For the determination of the air/fuel ratios of the individual cylinder the CO_2, CO, O_2 and N_2 contents in the exhaust leaving the trapping train and carbon/hydrogen ratio of the tested fuel were required. Under prevailing sampling conditions with an appreciable excess of oxygen assisting the combustion process it was reasonably assumed that all hydrogen in the fuel was combined with oxygen to form water. The formula was derived from the following considerations:

Take one pound mole of dry oxidized exhaust gas with excess oxygen in which CO_2, CO, O_2 anl N_2 represent the volume percentages of carbon dioxide, carbon monoxide, oxygen and nitrogen respectively; let N, A and F represent the weights of nitrogen, air and fuel in the corresponding quantity of unburnt mixture.

Then:

From the Orsat analysis of the outlet gas from the traps

$$N_2 = 100 - (CO_2 + CO + O_2)$$

Now $N = 28{\cdot}016\, N_2$

The corresponding quantity of air $A = \dfrac{100}{76{\cdot}32} N$

The weight of fuel associated with this air is

$$F = C + H = C\left(1 + \frac{H}{C}\right)$$

Where C and H are the weights of carbon and hydrogen in the unburnt fuel F.

But $C = 12{\cdot}01\,(CO_2 + CO)$

$$F = 12{\cdot}01\,(CO_2 + CO)\left(1 + \frac{H}{C}\right)$$

$\therefore$ The Air/Fuel Ratio A/F =

$$\frac{28{\cdot}016\,(100 - CO_2 - CO - O_2)}{12{\cdot}01\ \ (CO_2 + CO)(1 + H/C)} \times \frac{100}{76{\cdot}32}$$

$$3{\cdot}05 \times \frac{100 - (CO_2 + CO + O_2)}{(1 + H/C)(CO_2 + CO)}$$

The error caused by variations in carbon/hydrogen ratio of fuel reaching each cylinder, is normally negligible, mainly because of relatively small share of hydrogen in the molecular weight of fuel and of small differences in the carbon/hydrogen ratio of hydrocarbons forming the main body of normal motor gasolines.

If desired this source of error can be eliminated by measuring the relative quantities of carbon dioxide and water formed during the combustion.

Results of air/fuel ratio determined in each tested cylinder of the 6-cylinder engine operated on technical iso-octane +3 ml T.E.L/I.G. are given in Table 4.

During the work described the following test routine has been adopted:

The atmospheric conditions were recorded and operating temperatures, brake horsepower and specific fuel consumption were observed every 30 minutes. During the run, gas samples were taken at the trap outlet for Orsat analysis, and necessary adjustments were made to the oxygen supply and heater tube temperature if incomplete combustion was indicated by the presence of excess carbon monoxide. The normal sampling period used was 2½ hours.

At the conclusion of each test, the Pyrex precipitators and glass wool traps were removed and sealed, the weight of water in each trap determined, and the lead content of the deposits collected in the apparatus determined polarographically after solution in nitric acid. From the amount of lead collected and from the quantity of combustion water recovered from the condensing train the T.E.L. contents in fuel sampled from each individual cylinder were derived. Complete specimen calculations are given below.

1. *Lead Content.*

Tetra-ethyl lead, T.E.L.:—

Chemical Formula	Molecular Weight	Specific Gravity	Lead Content g Pb/g T.E.L.
$Pb\,(C_2H_5)_4$	323·45	1·659	0·64

Lead in 1 ml fuel:—

$\dfrac{0{\cdot}64 \times 1{\cdot}659}{4546} = 0{\cdot}000234$ g Pb/ml fuel with 1 ml T.E.L. per I.G.

$\therefore$ Lead content of leaded fuel:—

$0{\cdot}000234 \times \dfrac{\text{T.E.L. ml/I.G.}}{\text{s.gr. of fuel}}$ g Pb/g fuel

2. *Water/Fuel and Carbon/Hydrogen Ratios*:—

C—Carbon content of fuel, per cent.

H—Hydrogen content of fuel, per cent.

Water/fuel ratio $= 9 \times \dfrac{H}{100}$

Carbon/hydrogen ratio $= \dfrac{C}{H}$

3. *Humidity Effect.*

V—Measured total volume of sampled gas at N.T.P. (litres).

v—Total volume of unburnt mixture at N.T.P. (litres).

c—Volumetric contraction factor, taking into account condensation of exhaust water.

A/F—Air/fuel ratio by weight.

a—Total volume of air passed through the sampling orifice, at N.T.P. (litres).

f—Total volume of fuel sampled (litres).

γA—Specific gravity of air at N.T.P.

γF—Specific gravity of fuel.

h—Atmospheric humidity (grains of water per lb of dry air).

q—Weight of humidity water in total volume of air a (grams).

Volume of unburnt mixture $v=\frac{V}{c}$

For the range of air/fuel ratios measured $c=0{\cdot}92$ (mean value).

Air/fuel ratio by volume: $\frac{a}{f}=\frac{A}{F}\times\left(\frac{\gamma F}{\gamma A}\right)$

Total volume of air:—

$$a=\frac{a/f}{(a/f)+1}\times v \text{ litres.}$$

Weight of water in this volume of air:—

$$q=a\gamma A\frac{h}{7000} \text{ grams}$$

$$q=1{\cdot}847\times10^{-4} \text{ a.h. grams.}$$

4. *T.E.L. Recovery Percentage.*

W—Total water condensed in sampling train (grams).

q—Humidity water condensed in sampling train (grams).

L—Total lead recovered from sampling train (grams).

w—Water/fuel ratio.

l—Lead/fuel ratio.

Then fuel from lead recovered $=\frac{L}{l}$ grams.

Nett quantity of water obtained from combustion of fuel in water traps $=(W-q)$ grams.

Fuel calculated from water $=\frac{W-q}{w}$ grams.

and T.E.L. percentage recovery

$$=\frac{L}{l}\cdot\frac{w}{W-q}\cdot 100 \text{ per cent.}$$

$$=\frac{w}{l}\cdot\frac{L}{W-q}\cdot 100 \text{ per cent.}$$

Résumé

Examination of the bodies of the sampling units and heater tubes did not show any traces of deposited lead compounds after the tests; this indicates that all the lead which is associated with the sampled gas from the orifice is trapped in the apparatus. The deviations in the mean percentage of lead recovery from individual tests were maintained within approximately 5 per cent of the average lead recovery figure obtained from a given fuel and set engine running conditions. These, together with the similarity between T.E.L. distribution patterns for individual runs indicate that the sampling technique developed can be of useful service in the assessment of lead content in the charge to any individual cylinder of an engine.

The success of the technique is attributed to heating the orifice and the tubular passage so that lead compounds remain volatile until the sampled gas reaches the cooled surface of the glass precipitator on which the lead deposits, and from which it can conveniently be washed off for quantity determination. The second and equally important function of the tubular heater is to provide the ignition heat for the unburnt portion of the gaseous mixture drawn from the engine combustion chambers. This, together with oxygen feeding creates in the sampling unit conditions for the oxidation of combustible constituents in the sampled gas, thus making the determination of air/fuel ratio quite a simple operation. This type of treatment avoids the need, under rich mixture operation, for a comprehensive gravimetric analysis of the organic components in the exhaust gas. The lead recovery figures obtained indicate that approximately 40 per cent of the lead introduced in the fuel as T.E.L. cannot be found in the air/fuel mixture drawn from the combustion chambers of an engine. The average percentage of lead recovery from all cylinders of 59·3 is approximately 8 per cent higher than the mean lead recovery figures obtained from the engine exhaust gas in the early stages of the investigations. Downs[4] quotes for similar engine operating conditions a mean percentage lead recovery of 63·1, of which, however, only 51·2 per cent was obtained from the exhaust gas, the remaining 11·9 per cent being deposits accumulated in the combustion chamber during the test. In order to obtain some information concerning the whereabouts of the missing lead a complete lead balance test was carried out by Downs on the engine operating at 40 m.p.h. road conditions. The total percentage of lead recovered in this test was 81 of which 52·4 per cent was recovered from the exhaust gas, 20·5 per cent from deposits on all parts of the engine combustion chamber and 8·1 per cent from the lubricating oil. If a similar lead distribution is assumed for the test described, the percentage of lead unaccounted for will be 12·1. This loss may be, at least in part, due to underestimation of the lead left behind as deposits since it is known that deposits break away at random intervals and thus cannot be properly assessed for the lead balance.

It is considered that the continuous sampling technique as described is simpler than any other methods of equal accuracy since alternative techniques for obtaining similar information demand elaborate modifications to the engine and complicated trapping systems or as in exhaust or intake manifold sampling, leave doubt whether the samples obtained represent the composition of the actual air and fuel mixture in the engine cylinders.

The unavoidable departure from engineering simplicity in building up the apparatus is rewarded by the simple air/fuel ratio determination and by far less involved modifications to the engine.

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MECHANICAL HANDLING EXHIBITION

FROM June 9th to 19th, 1954, the fourth biennial Mechanical Handling Exhibition and Convention will be held at Olympia, London. More than 200 firms are exhibiting. All types of labour aids will be shown, many of them in operation.

SHIELDED ARC WELDING

Some Recent American Developments

FUSION welding is one of the most versatile and useful tools which modern industry has evolved. In countless applications it has replaced fastening by bolts and rivets and made possible new structural designs. This is particularly true in the high temperature products which the American Ryan Aeronautical Company builds for jet, rocket and piston engines. In these applications, the heat which must be endured requires that the components be as homogeneous as possible. Also, the supersonic flows of fluid through them demand smooth projection-free surfaces to avoid the incidence of turbulence and drag. "Inert gas-shielded arc welding" incorporates the advantages of other types of fusion welding and permits the joining of many high temperature "super alloys" which could not otherwise be fabricated.

There are five basic types of shielded arc welding in general use. These are: (1) Inert gas-shielded tungsten arc welding, popularly called "Heliarc," (2) Inert gas-shielded metal arc welding, called "Sigma," (3) Submerged arc welding, also called "Unionmelt," (4) Atomic hydrogen welding and (5) Coated electrode arc welding.

Each of these methods employs a shielding agent, either gas or solid, to protect the hot electrode and weld metals from the oxidizing effects of the atmosphere. While this oxidation can be tolerated in many industrial welding applications, it is thoroughly undesirable in others. For example, in jet, rocket and piston engine parts, oxidation produces a brittle, porous structure with reduced strength and corrosion resistance. As they are continually subjected to cyclic extremes in heating and cooling, these components must be joined by top quality welds. Similarly, aluminium and magnesium alloy structures must be shielded from oxidation when welded or the hard oxide film which forms will prevent the heating and joining of the metal.

Developed at the General Electric Company by Hobart and Devers, inert gas-shielded tungsten arc welding is similar to atomic hydrogen welding in that it makes use of a gas for protecting the hot metal. Unlike the atomic hydrogen process, the arc is maintained between a single tungsten electrode and the base metal instead of between two electrodes. Also, the blanketing gas, which flows in a low velocity stream around the electrode, consists of an inert, monatomic gas rather than hydrogen. This gas serves only as a shielding agent and does not dissociate to release additional heat as hydrogen does.

The Heliarc arc is quiet and the tungsten electrode consumption is extremely low. Filler rod may be added by simply feeding wire into the arc. Both A.C. and D.C. systems are used, depending upon the materials being joined. Argon and helium-shielded tungsten displays a marked polarity, or rectifying quality. When used with D.C. current, this causes unequal heating at the two ends of the arc. With straight polarity, the electrode operates cool and the work is hot, which is desirable for most metals. For use with aluminium, magnesium or beryllium-copper alloys, reverse polarity must be employed. In this the electrode is positive and operates hot with the work cooler.

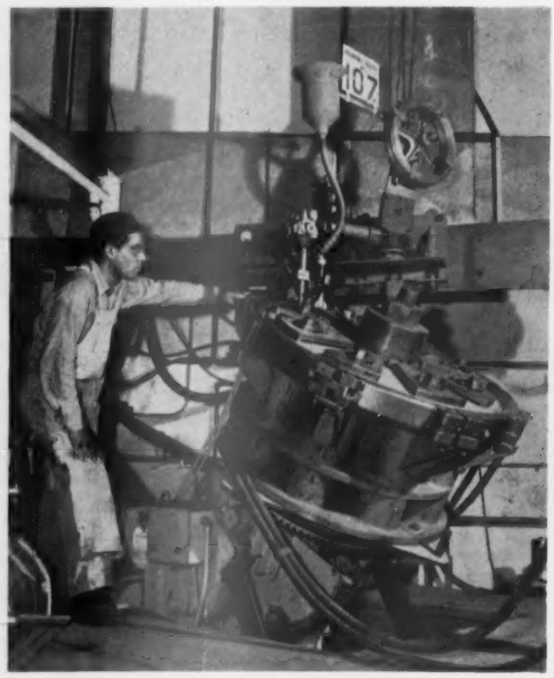
The outstanding advantages of Heliarc welding are elimination of flux, increased speed, improved quality and minimized weld finishing. By eliminating the need for flux, Heliarc provided the first successful means for fusion welding aluminium and magnesium alloys, which are attacked by the corrosive elements of fluxes. Without flux, flux entrapment is avoided. This permitted much more latitude in welding design because some joints are certain to entrap flux and often its removal is impossible. Ryan employs Heliarc to weld tee, lap and edge joints which were formerly regarded as impractical to attempt because of flux removal problems.

Heliarc is fast. Full penetration with speeds as high as 40 inches per minute can be attained on $\frac{1}{16}$ in stock. The pinpoint concentration of heat which Heliarc provides, coupled with its speed, makes it exceptionally useful for welding the stainless steels which the company fabricates. These alloys are subject to carbide precipitation and loss of corrosion resistance when exposed to sustained welding temperatures. Also, the stabilizing elements of columbium and titanium, contained in stabilized grades of stainless steel may be boiled away if metals are held at welding temperatures for prolonged periods.

Undoubtedly, Heliarc's narrow band and high speed heating characteristics have made possible the welding of the "super alloys" used in jet engines. Inconel X and W, Stellite 25 and N-155 are extremely susceptible to cracks at welding temperatures because of their poor thermal conductivity and high coefficients of expansion. Normal welding techniques pour heat into these alloys faster than they can conduct it away from the weld zone. As a result, extreme temperature differential is established between the hot and cold portions of the sheet and severe stresses are set up. Heliarc avoids these phenomena because of its speed and narrow zone of heating.

Heliarc is efficient for welding practically all the commercial metals including aluminium, magnesium, copper, titanium, stainless steels, nickel and cobalt base alloys, carbon steels, cast iron, lead and silver. It produces X-ray quality welds which possess good tensile strength, elongation and impact resistance. Its high speed makes it ideal for welding light gauge materials because it reduces warpage and provides attractive appearing welds. At Ryan's plant, Heliarc is used to weld gauge thicknesses running from 0·035 in to two sheets of 0·156 in.

Owing to its inherent characteristics and the stabilizing effect of the blanketing gas the Heliarc weld deposit is laid down in a smooth bead without sputter or splash. Consequently, the weld



A rotating fixture for welding jet engine burner plate by the Unionmelt process

seldom requires the grinding or finishing operations which are associated with other fusion welding operations.

The company rolls all Heliarc butt welds in austenitic steels, which are effected without the addition of filler wire, in seam rollers. Especially built for this work, the machines flatten the weld seam between two steel rolls under heavy pressure. This cold works the weld to refine its structure and eliminates the notch effect caused by the tendency of the metal to sag slightly when molten.

Because of the complex, irregular design of these exhaust systems, they are welded manually. Heliarc is well adapted to this work because it involves light weight equipment with some torches weighing only three ounces. For this work the company has established one of the most modern welding areas in the aircraft industry with 29 integrated Heliarc welding booths in one location. All motor-generators are installed overhead and a cascade system is used to supply argon gas and cooling water. Gas and water are automatically controlled by modern timers.

For these tasks, D.C. straight polarity current, ranging from 75 to 100 amperes, is used. Material thicknesses vary in the designs and the Ryan welder uses a Mullenbach foot control rheostat to vary the welding voltage as he goes. This allows the welder to attain better control of welding conditions and produce a finer weld result. Instead of a filler metal being added to these welds, an upstanding flange is formed at the seams and melted down by the heat of the Heliarc arc.

In addition to high temperature alloys, the company welds many aluminium alloys with manually controlled Heliarc. Typical example of this work is the welding of huge fuel tank rings, 4 feet in diameter. These rings serve as splice joints for the largest external fuel tanks ever built for aircraft. They are fabricated from 61S aluminium alloy. Ryan welds them with high frequency stabilized A.C. current, using between 75 and 150 amperes, depending upon thicknesses encountered. These gauges vary from 0·050 in to 0·072 in. Heliarc torches used are exactly the same as those employed in welding stainless steels.

One of the interesting types of welding which only Heliarc is fitted to effect is fusion spot welding. With a special Linde-pistol type torch, Heliarc can be used to make automatically-timed spot welds which are exceptionally useful in certain production jobs. Because the spot welds can be accomplished from only one side of the structure, the parts do not need to be removed from jigs but can be spot welded right in their fixtures. The technique is valuable where several components must be aligned in a precision fixture and spot welded in place.

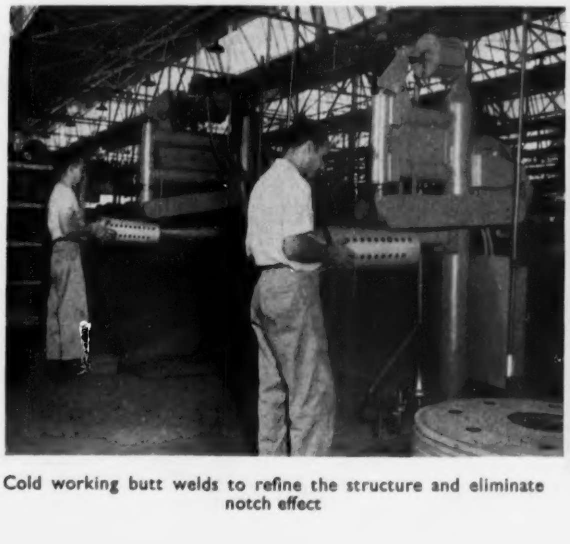
A mobile machine incorporates all of the necessary elements for Heliarc spot welding—D.C. generators, argon gas supply, 40-gallon water tank and pump. With this device, spot welding facility can be quickly moved along the production lines to the work. The new machine is used to spot weld Fairchild C-119B exhaust hoods and machined rings in assembly jigs. After these parts are carefully fitted together and spot welded, they are seam welded in resistance welding machines. The Heliarc torch produces such a smooth spot weld that no finishing is necessary in order to run these parts between the wheel electrodes of the resistance welder.

In addition to the large amounts of manual Heliarc welding, substantial quantities of automatic Heliarc are accomplished at the plant. Automatic machines are used on all straight line, butt welding jobs, both in aluminium alloys and the stainless steels. For example, the Company builds afterburners, tail-pipes and exhaust cones for General Electric J-47 jet engines in this way. These structures are fabricated from stainless steels and other high temperature alloys in thicknesses from 0·040 in to 0·075 in. The company designs and builds special fixtures to hold the parts in exact alignment while the automatic Heliarc welding head traverses their longitudinal or circumferential seams. D.C. straight polarity current is used and welding speeds up to 30 inches per minute are attained. A number of machines are equipped with selenium-type rectifiers which convert A.C. current from the line into D.C. and eliminate the motor generator.

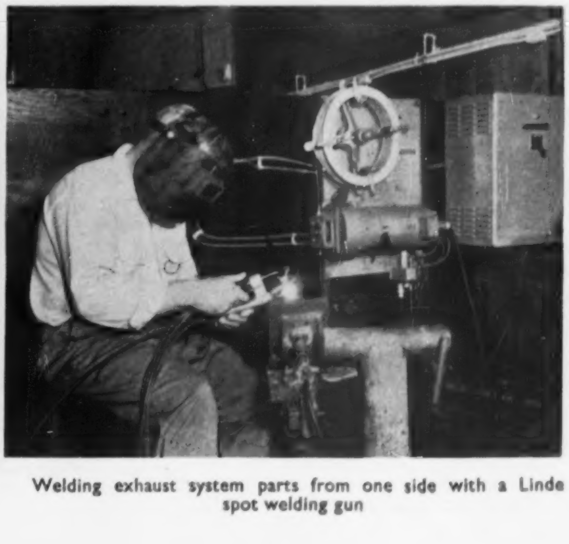
All the longitudinal seams in huge fuel tanks are welded with automatic Heliarc machines. These large sections are butt-welded with filler wire automatically fed into the seam. The alloy is 61S aluminium and gauge thicknesses run from 0·051 in to 0·072 in. With the Airco "floating" head which automatically maintains a constant length arc, it is possible to weld contoured surfaces. The company has built such a machine and uses it to weld automatically the irregular contours of afterburner nozzles.

One of the newest devices, which Ryan is using, is the Linde semi-automatic HWM-1 welder. This is a manually operated Heliarc torch which automatically feeds filler wire into the arc. All the operator has to do is set the current and wire feed and guide the work. When the arc is struck, the wire is automatically fed between rolls into the weld. Its movement propels the torch. Hand welding speeds as high as 80 inches per minute have been reached with this new method. Combining the advantages of manual guidance with automatic control of arc length, wire feed and travel, the technique is excellent for irregular components. It is water-cooled and capable of handling up to 300 amperes capacity.

A great deal more argon gas than helium is being used in connection with Heliarc welding. Basically, this is because argon can be obtained in higher purity commercially and is more efficient for most of the Heliarc welding tasks. Ten times heavier than helium, argon flows down over the welding better and much less is required to

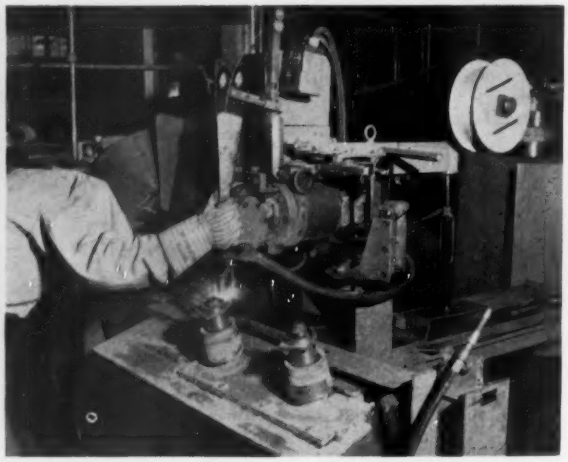


Cold working butt welds to refine the structure and eliminate notch effect



Welding exhaust system parts from one side with a Linde spot welding gun

Automatic Heliarc machine welding stainless steel at 30 in per minute



An automatic machine for Sigma welding exhaust system nipples in seven seconds

perform a similar welding assignment. Argon arcs have lower voltages than helium arcs and thus permit the use of higher currents without burning the work. Both lower voltage and increased current contribute to improved arc stability. The company does use helium, mixed with argon, for all welding of aluminium alloys because the heat of the helium arc is greater for a given current than that of argon and penetration is deeper.

For speed in welding, inert gas-shielded metal arc—Sigma—makes other types look slow. This amazing method has been clocked at 200 inches a minute with 1/16 in material. Even in 1/4 in plate, structural welds can be made at 48 inches per minute. Sigma uses a consumable wire electrode which is fed at high speed through a blanket of argon gas and melted into the weld area. Like Heliarc, it produces high quality welds with deep penetration and lends itself to automatic operation. Sigma welding can handle twice the current densities which are used with Heliarc.

The company uses Sigma to weld exhaust system nipples to port flanges and join ball sections to exhaust tubes. These components are about two-and-a-half inches in diameter and are welded circumferentially. For welding the nipples to the flanges, the company has devised a new machine which is outstanding in effectiveness.

The welding unit used is a Linde FSM-1 type powered by a Lincoln 400-A motor-generator. The unit is mounted on a custom-built fixture which incorporates a sliding carriage with two revolving holders that turn at constant speed. Each holder is a small expanding mandrel which firmly retains the nipple during welding and quickly releases it when contracted. The carriage moves back and forth through an 18-inch travel to bring the nipples directly under the electrode of the Sigma welder. It is operated by foot pedal control.

The operational sequence goes like this: the operator places a nipple on one of the revolving holders and presses the foot pedal. The carriage shifts to bring the nipple under the electrode. As it moves, the carriage trips a micro-switch which starts the welding cycle and the arc strikes. Current used is 250 amperes, reverse polarity D.C. The light of the arc plays upon a photo-electric cell which trips a relay that automatically times the welding run and switches off the arc. All the operator has to do is shift the carriage and replace the nipples. With two holders, one nipple is being welded while the other one is being replaced by the operator.

Formerly it required three minutes of manual welding plus finishing to do one of these nipples. With the new machine it takes only 7 seconds to attain a sound, clean weld which requires no cleaning or finishing labour. A normal production for this machine is 800 nipples per 8-hour shift although as many as 1,200 have been accomplished in that period. The company's own exhaust system production uses thousands of these stainless steel components and it is quite evident that the new machine has substantially reduced their cost and increased their availability.

APRIL ISSUE

The April issue of this journal will include a review of the exhibits at the forthcoming **Geneva Show**. *In the main, the review will deal with passenger cars, with particular attention to technical developments on vehicles that are not normally exhibited at the London Show. Continental body styling will be fully reviewed. This issue will allow those who cannot visit the show to keep abreast of European developments.*

Also in the April issue there will be a complete technical description of a new fully-automatic plating plant that is outstanding for size, for conveying equipment that is employed and for the quality of the product that is obtained.

The submerged arc welding technique—Unionmelt—is similar to Sigma welding in that it also employs a consumable electrode and permits the use of high current densities. It is different in that the shielding agent is not a gas but consists of dry, granular flux which pours down over the weld area. The arc and weld metal are fully protected from oxidation by this means. The granular flux melts and forms a fused slag which blankets the weld. The material which does not melt is recaptured and used again. The solidified slag is easily removed from the weld.

With Unionmelt there is no arc visible and the operation is quiet and smooth. Because Unionmelt permits the use of higher density currents much larger diameter electrodes can be used and heavier welds laid down in single passes. For this reason, it is used to weld heavy gauge materials and lends itself to straightaway work. Speed is a salient feature of Unionmelt. Automatic machines can weld metals three inches thick in one pass and can attain 200 inches per minute on light gauge materials.

Ryan uses automatic Unionmelt to weld a large heavy-gauge circular burner plate and flange together. This assembly is part of the aft frame, or "backbone" of General Electric J-47 jet engines. The big burner plate is 0·125 in thick and the flange is 0·075 in. Both components are fabricated from 321 stainless steel. The work is performed on a specially-built fixture which is rotated by means of a large Worthington weld positioner. Filler wire of 29-9 composition is added to the weld. Speeds of welding are 55 inches per minute to produce a 1/4 in fillet weld in one pass. Unionmelt is employed to weld stainless steel, carbon steels, Monel, Inconel, aluminium-bronze, copper-silicon alloys and straight chrome and nickel. Smooth, sound welds can be made at good speeds with 20 pounds of metal per hour deposited.

GUY ARAB BUS CHASSIS

A Six-wheeled Development for Overseas Service

A RIGID, six-wheeled single-deck bus chassis in the well-known "Arab" range has been developed by Guy Motors Ltd., Fallings Park, Wolverhampton, for service in Rhodesia. It incorporates a Gardner 6HLW underfloor engine, and a fluid coupling, pre-selective, epicyclic gearbox. The principal dimensions are:—

Overall length of chassis	34 ft $2\frac{3}{8}$ in
Overall width of chassis	7 ft $11\frac{3}{4}$ in
Wheelbase	18 ft 4 in
Front overhang ..	6 ft $10\frac{3}{8}$ in
Rear overhang	9 ft 0 in
Front track	6 ft $10\frac{9}{16}$ in
Rear track	6 ft $6\frac{1}{2}$ in

Engine

A Gardner 6HLW type engine is employed. It is specially designed for underfloor installation in a horizontal position. This six-cylinder unit is of the direct injection type, with $4\frac{1}{4}$ in bore, 6 in stroke and 8·4 litres capacity. The compression ratio is 13:1. When set for sea-level operation, the engine has a maximum output of 112 b.h.p. at a governed speed of 1,700 r.p.m. The maximum torque is 358 lb-ft at 1,300 r.p.m.

The cylinder block comprises two three-cylinder units cast in special ductile, high tensile iron. Renewable, dry liners of chromium alloy cast iron are fitted. Each unit is bolted to the crankcase by means of high tensile through bolts. These bolts also form the main bearing cap studs. This gives a very rigid construction and relieves the crankcase of much load, so that the possibility of distortion is greatly reduced. The crankcase is a high purity aluminium alloy casting. Both the sump and the timing case cover are aluminium castings, and each is finned at the front for cooling purposes.

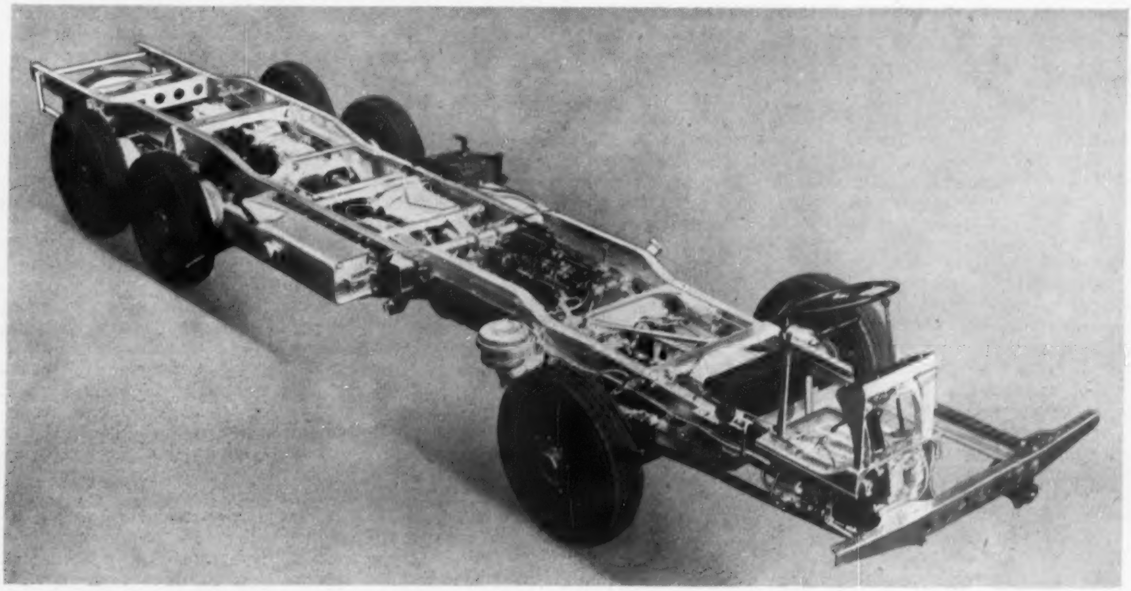
High tensile cast iron is used for the cylinder heads, which are fitted in two three-cylinder units. They have pressed-in valve seat inserts of special purpose, hardened alloy iron, which are highly resistant to wear and deformation. The heads incorporate special cylinder decompression gear to facilitate turning the engine during servicing adjustments. The mechanism slightly lifts all inlet valves. It is adjustable by means of a small, accessible hand-operating lever on each cylinder head.

For engine cooling, water is circulated through generous passages by means of a centrifugal-type pump that is positively driven from the engine camshaft at crankshaft speed. This unit is fitted with a spherical carbon gland in conjunction with a self-aligning ball race carrying the bronze impeller spindle, an arrangement that provides an exceptionally long wearing and efficient watertight seal. Including the radiator, the total water capacity of the cooling system is approximately eight gallons.

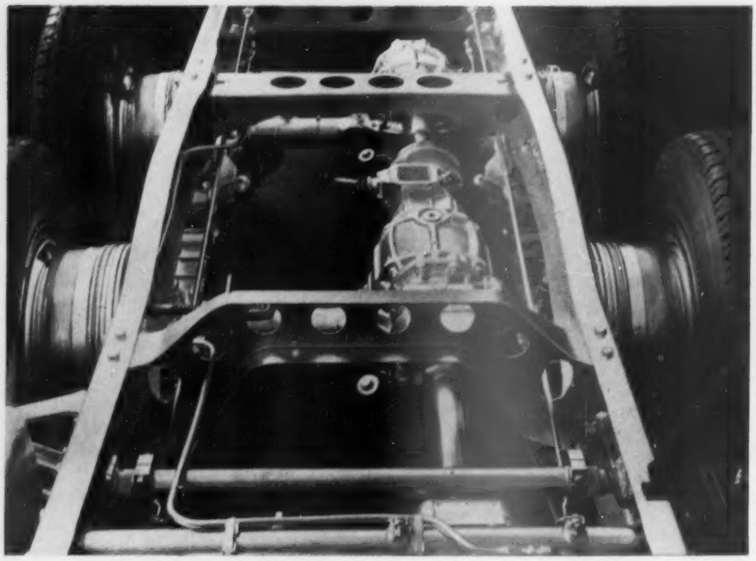
The crankshaft is a high tensile, chrome-molybdenum steel forging. It runs in seven white metal-lined bearings with bronze shells. Natural torsional oscillations of the assembly are controlled by an internal, friction-type vibration damper located at the forward end of the crankshaft. The valve camshaft is a steel forging carried in white metal-lined bearings in the crankcase. Low expansion aluminium alloy pistons are fitted. Each has three pressure and one scraper rings and the two top rings are chromium plated. Fully-floating, $1\frac{5}{8}$ in diameter, gudgeon pins are employed. They are located by aluminium pads fitted into the ends.

Engine lubrication is effected by means of a circulation-pressure system maintained by a gear pump that is driven from the valve camshaft. This is augmented by a scavenger pump driven in tandem with the pressure pump. Primary filtration is by means of an internal and detachable gauze element, suction strainer, which is mounted at the base of the pump and embodies a sediment trap. From the circulation pump, oil is fed through a passage in the crankcase to a pressure regulation valve and thence to a second filter. After passing the second filter the oil is delivered to the main bearings and the rocker mechanism. Surplus oil rejected by the regulator valve is piped to the governor unit, the injection pump cams and tappets and the main timing drive.

Except for a small portion of the sump, containing the oil filler, oil delivery filter and the dipstick, the engine is contained within the width of the frame. Three-point mounting is employed. There is twin rubber-



A rigid six-wheeled Guy single-deck bus chassis for service in Rhodesia



Final drive arrangement for the two rear axles. The forward of these axles incorporates a third differential that can be locked

bushed front link suspension from a support beam and bracket bolted to a channel section cross member. At the rear there is single point suspension from a tubular cross member, the engine being supported by a composite "G" shaped bracket with a bolt attachment that is fully cushioned by a rubber bush and horizontal rubber blocks. Engine movement is controlled and adjusted by a rubber-bushed torque reaction member. In addition a small hydraulic cylinder stabilizes the engine against the effects of exceptional cold starting conditions and when stopping. The engine also has an adjustable locating link at the rear for resisting fore and aft movement.

If desired a complete overhaul of the power unit can be effected *in situ*, since the cylinder heads, valves, etc., are easily accessible through hinged valances in the body skirting on the right-hand side, and big end bearings, main bearings, etc., from the left-hand side. In addition, the engine is easily removable from the chassis by means of a special three-point jacking system that can be supplied if required.

Air for the engine is drawn in through a large, heavy duty, oil bath cleaner. This unit is independently mounted on the right-hand chassis side member. It has a large diameter, corrugated rubber hose connection to the engine induction manifold. The intake from the filter is ducted to an air intake grille that is high up in the body side to ensure a clean supply of air as free as possible from road dust thrown up by the wheels. A baffle is fitted in the air intake to prevent entry of water when the vehicle is being washed down.

Fuel injection

A C.A.V/Gardner type B.P.F. flange-mounted fuel pump is used. The complete assembly is fitted in two three-pump units and is operated by means of Gardner cams and spring-loaded tappets. Each fuel ram has a hand operating lever for priming the system and testing the nozzles. The fuel camshaft is carried in large diameter ball bearings and is driven through a helical gear that meshes with a helical gear on the valve camshaft. The fuel camshaft gear is free to slide on a helical spline, and the slight axial movement provides a corresponding change of phase between the pump shaft and the driving shaft, thereby varying the injection timing. By means of a yoke coupled to the accelerating lever, the gear is moved axially as the accelerator is operated, and thus the timing is varied automatically with the engine speed.

From the tank, fuel is fed through a frame-mounted primary filter by means of a diaphragm-type lift pump. This pump is mounted on top of the engine crankcase and is operated by an eccentric on the valve camshaft. It delivers fuel to a combined overflow and separator unit that permits the escape of any air which may have been drawn into the pipe line at some point on the suction side. At the same time, a suitable fuel head is delivered to the injection pumps via a second filter, while excess fuel is piped back to the fuel tank. Both fuel filters are of special design. Each has two gauze elements, an inner and an outer, which are easily removable for cleaning.

A centrifugally-weighted "all-speed" governor unit is mounted on an extension of the fuel pump camshaft. It maintains close control and gives immediate response between idling and maximum speeds in all gears. A special cold-starting device is incorporated. It is actuated by means of a spring-loaded control handle on the right-hand frame side member. The solenoid-operated engine stopping control is connected to a separate lever mounted on the governor housing. This lever acts directly on the injection pump control rack and is independent of the accelerator control linkage.

Gearbox

The transmission includes a fluid coupling and pre-selective gearbox. As the engine crankshaft centre-line is offset towards the left-hand side of the frame, the gearbox and overhead worm type final drive are similarly offset to maintain a straight transmission line. The gearbox is three-point independently mounted between two large diameter tubular cross members. At the front, the mounting bracket is suspended from a pair of saddle brackets. Each of these saddle brackets is secured to the cross member by an inverted U bolt. A ¾ in. diameter, high steel suspension bolt locates the gearbox bracket between the saddles.

At the rear, the gearbox off-side and near-side mounting brackets are suspended from a pair of eye bolts. These bolts pass through the tubular cross member and are provided with top and bottom locating saddles. The gearbox bracket locating bosses are forked to receive the eye bolts and the attachment is completed by ⅝ in diameter, horizontal, high tensile steel suspension bolts.

Drive from the engine to the gearbox is through a short tubular propeller shaft. The gearbox provides five forward and a reverse speeds, and the ratios are:—

1st	...... 5·5 : 1
2nd	...... 4·2 : 1
3rd	...... 2·38 : 1
4th	...... 1·57 : 1
5th	...... Direct
Reverse	.. 9·09 : 1.

Top gear is obtained through a multi-plate clutch. Five trains of epicyclic gears are interconnected and compounded to provide the other four forward speeds and the reverse.

Gear engagement is air operated. It is effected by full depression and release of the pedal, which is mounted on needle bearings and provided with a renewable bonded rubber-to-metal wearing pad. The pedal is connected directly to the air control valve, which in turn controls the air pressure in the gearbox power cylinder. When the pedal is depressed, the air pressure is temporarily released by the control valve and change of gear is effected. Directly the pedal is released, the control valve allows the air supply to build up pressure in the gearbox power cylinder and hold the brake band in contact with the selected gear. At the same time the control valve acts as a reducer to ensure that pressure in the gearbox cylinder does not exceed the predetermined maximum.

Gears are pre-selected by means of a normal, ball type lever that operates in an "H" type gate. The selector lever engages slots in three shaft mounted rocking gears which, in turn, mesh with a cluster segment on the operating

shaft. Variable movement of the operating shaft lever depends upon which pair of segments is engaged. The shaft passes through the side of the control box and has a serrated end to receive the operating lever.

The operating lever is connected to the selector camshaft lever on the gearbox by means of a cross-shaft at the front of the chassis and a neatly arranged linkage adjacent to the left-hand side member of the chassis. The cross shaft is carried in double-row, self-aligning ball bearings housed in composite, cast aluminium brackets bolted to the main frame structure. All operating rod relay levers are mounted on needle bearings and have steel wearing washers. The relay levers have phosphor-bronze bushes in the eyes to take the operating rod jaw pins.

Bushes and rotating parts in the gearbox are lubricated by oil delivered through the centre of the driving shaft by an eccentrically driven plunger operating in an oscillating cylinder housed at the forward end of the gearbox.

Double axle driving bogie

Both axles of the double axle driving bogie have overhead worm drive with an axle reduction ratio of 6·25 : 1. The forward axle of this assembly incorporates a third differential that can be locked by means of an operating lever mounted on top of the worm casing. This lock is intended for use when the road conditions are very greasy or when the vehicle is ditched. Under all normal running conditions the third differential is left in operation.

The rear bogie is of the balancing beam type in which each axle is mounted on two semi-elliptic springs which are shackled to balance beams at their ends. The balancing beams are supported from two frame brackets that are rigidly braced by means of a channel section cross member running across the frame at the centre-line of the bogie. Bonded rubber bushes are used in the balancing beam to eliminate the need for lubrication.

Fully floating axle shafts forged from alloy steel are housed in a one-piece axle casing. The five-start, 8 in centres overhead worm, housed in a cast aluminium casing, is offset to the nearside of the chassis to give minimum floor height. The worm shafts are carried in double taper roller bearings at the rear ends and in roller journal bearings at the front to allow for expansion. Long-wearing face type oil seals are incorporated in the assembly. Shims are employed to provide correct worm shaft end float in the rear bearing.

Twelve ¾ in diameter studs are used in the assembly of the "four star" differential units. These units are carried in taper roller bearings, and the bevel wheels and pinions have renewable hardened thrust washers that are positively located to prevent rotation on the differential yokes. The rear hubs are mounted on taper roller bearings with spacers and shims between the inner races.

Synthetic oil seals are fitted at either side of the differential bearings and there are secondary seals at the outside of each hub outer bearing. A seal is also fitted at the hub inner bearing position to prevent the escape of hub grease on to the brake linings. As a further precautionary measure, oil collectors are fitted inside the brake drum. Excessive axle movement is prevented by means of rubber bumping blocks fitted at each side of the axles. An oil filler spout that gives visual indication of the correct oil level is fitted to each axle. The oil capacity is 2½ gallons.

Front axle

A reversed Elliot type front axle is fitted. It has an "I" section beam of high carbon alloy steel. The "I" section is retained between the spring pad and the king pin boss in order to provide maximum resistance to braking torque. Case-hardened chromium plated king pins are fitted. They have taper-roller thrust bearings at the top and phosphor-bronze bushes at the bottom. Provision is made for individual lubrication.

The hubs are carried on taper roller bearings, with spacers and shims between the inner races to provide simple means of hub adjustment. Oil seals are fitted at the hub inner bearings to prevent escape of hub grease on to the brake linings, and as an additional precaution oil catchers are fitted inside the brake drums. The brake anchor pins are case-hardened and chromium plated.

Frame

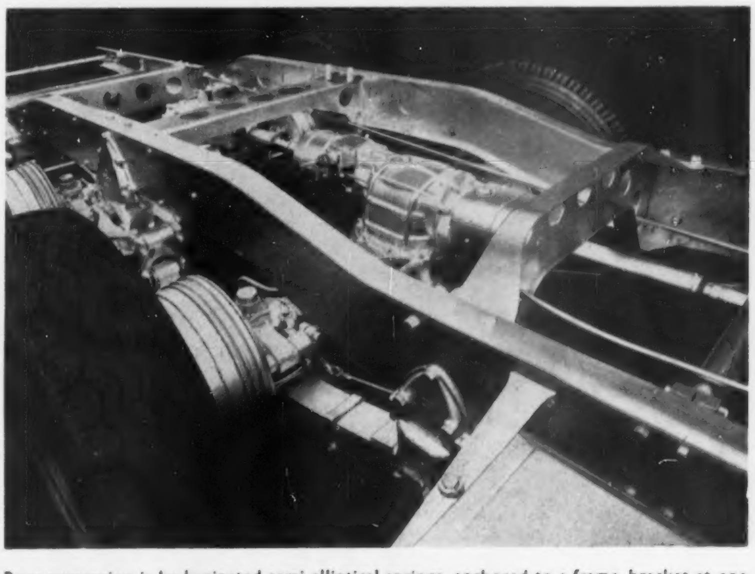
Bolted construction is used throughout the frame assembly. The main frame members are made from channel section nickel steel ¼ in thick and with a maximum section 10 in deep × 2⅞ in wide. At points of maximum stress, channel section flitch plates are inserted to give added strength and rigidity. The ends of these reinforcing channels are suitably tapered off to obviate any sudden change of section, and consequent stress concentration, in the assembly.

Cross bracing is effected by means of six pressed steel channel section members and three large diameter tubular members, and there is also a 2 in diameter tail stiffener. The foremost cross member is extended outside the frame to afford full width support for front body mounting. To provide a high degree of accessibility to components, the frame is swept up over the power unit. A towing loop is fitted to the underside of the front cross member. The chassis incorporates a "straight" frame rear overhang for use with the standard bus body.

Suspension

At both the front and rear, suspension is by six heavy-duty laminated semi-elliptic springs made from silico-manganese steel. All the springs are 4 in wide and the centre distances are: front 62 in, rear 48¾ in. Each spring assembly has an anti-roll device at each end to provide comfortable riding combined with good stability.

Each rear axle is mounted on two under-slung semi-elliptic springs, with each spring anchored to a frame bracket at one end and shackled to a balance beam at the other end. The main leaf of each spring has a machined tension surface and incorporates substantially strengthened rolled eyes and bonded rubber-to-metal shackle pins to eliminate the need for lubrication. Furthermore, the main leaf of each spring is reinforced by an extra rebound leaf and the tension sides and edges of all leaves are shot-blasted. Spring anchor brackets are a feature of the suspension. These are composite units



Rear suspension is by laminated semi-elliptical springs, anchored to a frame bracket at one end and shackled to a balance beam at the other

and are interchangeable. Overload buffers are provided on both front and rear mountings. Direct acting, telescopic type, hydraulic shock absorbers are fitted to the front axle.

Brakes

Air pressure brakes, pedal-operated from power cylinders, are fitted on all six wheels. A twin-cylinder air compressor is mounted inside the chassis frame. It is driven by twin vee belts from the gearbox front coupling, and has rack adjustment for the maintenance of the correct belt tension. The front power cylinders are mounted over the axle swivels and operate directly on the brake camshaft levers.

The four rear power cylinders are mounted on the axle casings. They also operate directly on the respective brake camshaft levers. The compressed air reservoir unit is mounted on the near-side chassis side member. It incorporates an unloader valve, which relieves the compressor of most of the pumping load when the reservoir is charged to operating pressure.

All brake linings are ½ in thick. Those at the front are 16½ in diameter by 6½ in wide and those at the rear 16½ in diameter by 6½ in wide. This gives a total braking area of 935 in^2. The brake drums are made from high grade cast iron. They all have deep, wide section flanges to prevent distortion. The rear drums are also ribbed to assist heat dissipation. Furthermore, to ensure efficient cooling, the rear drums are extended beyond the wheels on the inner side. Three-pawl type, automatic shoe adjusters are fitted at all six wheels. They allow the full wearing life to be obtained from the linings. Anti-squeal bands are fitted to all brake drums.

The hand brake lever is a one-piece forging of the "pull-on" type. It has a chromium-plated trigger and grip. In a conventional rack and pawl mechanism, maximum wear of the hand-brake pawl occurs when the lever is in the "off" position. On this chassis an extremely rigid mounting has been developed in which a double-toothed trailing pawl is so arranged as to be out of mesh with the quadrant when the hand brake is in the "off" position. This mechanism and the handbrake lever pivot and accompanying needle bearings are totally enclosed in a cast aluminium housing that provides positive draught-proofing and adequate protection against mud and water. To facilitate maintenance, the housing is fitted with an easily detachable cover. The hand brake linkage is purely mechanical and independent of the footbrake. Operation is on the rear wheels only and on the same shoes as the footbrake. The hand brake relay cross-shaft is mounted on double-row self-aligning ball bearings.

Radiator

A contra-flow type radiator is employed. It has detachable cast aluminium tanks internally treated with bituminous anti-corrosive paint and externally with dull black heat-resistant paint. The unit has an ample cooling surface of 50 horizontal Still tubes. It is vertically mounted on rubber cushions below floor level. To eliminate the possibility of fouling by road deposits, the unit is positioned ahead of the front axle.

A 20 in diameter, six-blade, closely shrouded axial-flow fan unit is housed in a cast aluminium cowl. The cowl is bolted to the radiator at the front and single-point mounted at the rear by twin swinging links with rubber bushes. Drive to the fan is through a jackshaft that is connected by flexible couplings to the front end of the crankshaft. The driving pulley is mounted on the forward end of the jackshaft and triple vee belts transmit the drive to the fan spindle pulley. To permit correct belt tension to be maintained, the driving pulley bearing housing is eccentrically mounted.

Cam and double roller type steering, with a 28·5 : 1 ratio, is employed. It gives smooth and easy steering with a light castor return action. The steering wheel is 21 in diameter. Spring-loaded ball joints, which are self-adjusting, are incorporated in the track rod and drag link assembly. A moulded rubber, oil resisting cover is fitted to both the front and rear of the drag link to prevent the entry of dirt. Adequate steering stops are fitted to the front axle ends to prevent shock from being taken in the steering box.

EXHAUST VALVE MATERIALS

An article, "Some Significant Properties of Exhaust Valve Materials," by J. L. Danis, has been published in *The Eaton Forum*, Vol. 14, No. 1. The object of the article is to present, as a guide to material selection, some of the important properties of valve materials. Physical and chemical data obtained in the laboratory admittedly require follow-up testing on the dynamometer and in the field, but such data assist in determining which are unsuitable materials.

An important property of exhaust valve steels is their hardness at elevated temperatures, and it is stressed that, if an indenter method is used, the indentation time should be constant and the indenter itself should be hot. A better method is that of mutual indentation testing, which avoids difficulties arising because of the need to take into account the hot-hardness of the indenter. It is essential that the conditions of fabrication and heat-treatment normally used in the manufacture of the valves should be applied to the preparation of the specimen for hot-hardness testing. A chart is given showing the hot-hardness values, between 600 and 1,600 deg F, of some ten materials, including certain important ones used for seat-facing.

One property common to all exhaust valve steels is the development of a continuous, stable and adherent protective oxide layer at temperatures below 1,500 to 1,600 deg F. These layers are variously affected by vapour and finely divided powder contaminants, and some indication of the reaction between the oxide and such contaminants may be gained by a lead-oxide sticking test described. Powdered lead-oxide is heaped on pre-oxidized specimens, and the temperatures are noted at which: (a) the oxide attaches itself to the valve-steel oxide, and (b) the heap fuses to dissolve away the protective film and expose a bare reactive metal surface.

Many valve breakages occur by rupture, stress corrosion or corrosion fatigue in that section which operates in the neighbourhood of 1250 to 1500 deg F. A convenient index for assessing resistance to this type of breakage is the stress required to produce 1 per cent total stretch in 10 hours at 1350 deg F. Heat-treated En 62 was the most stretch-resistant of the more common austenitic valve steels, but it was found that various Inconel types, which may well receive more attention in the future, were even better. *M.I.R.A. Abstract No.* 6333.

TEMPERATURE MEASUREMENT

A NEW type of pyrometer has been developed, which might be applied to reciprocating type, internal combustion engines. This instrument was initially designed for the measurement of temperatures of fast-moving components in gas turbines, and it was described in the August 1953 issue of *The Engineers' Digest.* A concave mirror system is employed in conjunction with a pyrometer, and it collects and reflects the heat radiation received from the component under observation. Although in the turbine application the gas temperature was low at the point where the gases reach the mirror, it was nevertheless high enough to tarnish ordinary mirrors. Rhodium plated mirrors, reflecting heat rays on a special type of thermo-element, were finally found to give satisfactory service. The thermo-element consists of 30 small thermocouples arranged in a series within a circle of 5 mm diameter, and it is stated that it is capable of measuring temperatures of up to 800 deg C, with an accuracy of ±5 deg C.

ENGINE RESEARCH

The Laboratories of the British Internal Combustion Engine Research Association

THE British Internal Combustion Engine Research Association, generally abbreviated to BICERA, have large and exceptionally well-equipped laboratories at Slough, Bucks. Their work is of a rather more specialized nature than that of many similar co-operative research organizations, so they can to a greater extent concentrate their efforts instead of having to cover a wide range of problems. Even when dealing only with internal combustion engines, the scope for research is considerable, but there are many problems that are common to the whole range of petrol and diesel units, from the smallest to the largest. This is important in that the findings are often of immediate practical use and interest to the majority of members. Much of the work of the Association is, however, concerned solely with diesel engines, particularly those for industrial and auxiliary marine applications.

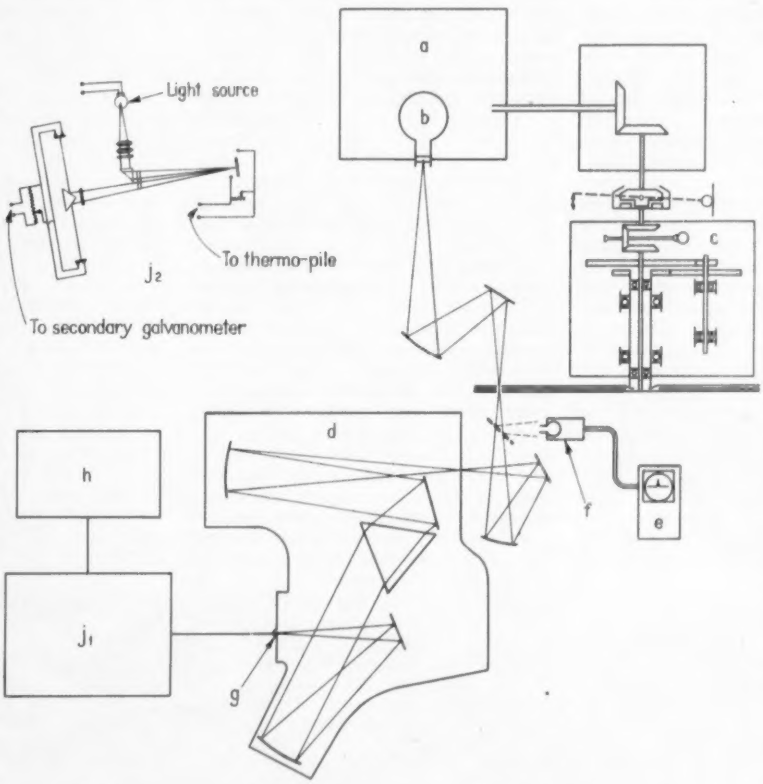
That the considerable expenditure involved in equipping the laboratories to their present standard has been fully justified was demonstrated recently when the laboratories were open to visitors for two days. This demonstration showed that in addition to fundamental research being done, components of entirely original design are being developed for use on internal combustion engines, and improved design methods are being perfected. Some of these methods are already being applied in practice by members of the Association. They include a means of estimating crankshaft torsional stiffness to within about +3 to −6 per cent accuracy, and a new procedure for the design of viscous fluid dampers. Among the new components developed are an improved fuel injection pump actuated hydraulically, a delivery valve for eliminating secondary injection, and which may be used for pilot injection, a piston giving an automatically variable compression ratio, and the Bicera rotary compressor.

Low grade fuels

Economic benefits have been obtained by the use of boiler fuels in large marine diesel engines, and this has led to an investigation to determine whether or not these less expensive fuels can be used in smaller engines. The doubt as to whether they can be used arises because, whereas in bigger engines a relatively large amount of wear represents only a small percentage of the cylinder bore, the same amount of wear in small units is a correspondingly large percentage of the bore. Furthermore, in smaller units, there is a greater tendency for injectors, valves and rings to become fouled. The fuels that operators would like to use are those that meet the Class B requirements of B.S.209 but fail to meet the Class A requirements, usually in regard to sulphur content and Conradson carbon value.

To investigate this problem systematically, an assured supply of low-grade fuel of consistent quality was required. Moreover, this fuel had to possess, so far as possible, all the worst properties that can be found in fuels of the Class B category. For the purpose of the tests, the Shell Petroleum Co. Ltd. made available a reference fuel, designated DR 63. Although this fuel did not incorporate all the worst possible features, for this would be impossible, it was considered that an engine that would run satisfactorily for long periods with this fuel would be certain to give good service with any commercially available Class B fuel.

The engines used for the tests were a Tangye VC1 unit, of 4·5 in bore, a Perkins P6V engine, of 3·5 in bore, and six Petter AV1 series II engines, of 3·15 in bore. Both straight and detergent lubricating oils were employed. During the tests it was found that the most serious troubles that occurred when the engines were run with the reference fuel were severe pitting of the inlet and exhaust valves and seats, and excessive wear of the piston rings, ring grooves and cylinder bores. Undue scuffing of the top piston land was also experienced, particularly on the smaller engines. In all the earlier tests, heavy deposits of carbon were found and the lubricating oil became rapidly contaminated. As a result of this work, which



a Engine
b Sapphire window
c Strobometer
d Spectrometer
e Oscillograph
f Photo-cell for indicating opening of strobometer shutter
g Thermopile
h Secondary galvanometer
j_1 Galvanometer and optical amplifier
j_2 Layout of the optical amplifier in j_1

Fig. 1. A high speed strobometer is employed in conjunction with a spectrometer to study combustion

involved more than 25,000 hours of engine testing, improvements were effected which, in general, reduce the rates of wear in the ratio of about 5:1. Even so, these rates are still somewhat greater than those obtained with Class A fuels. The findings have enabled recommendations to be made regarding not only the design but also the maintenance of engines in which the use of low grade fuels is to be allowed.

Recent investigations have tended to indicate that in power units running on lower grade fuels, particularly those which run at more or less constant temperatures for long periods, most of the wear of the rings, grooves and cylinder bores is due to the abrasive action of hard carbon particles. For this reason, it was thought that a detailed study of the combustion process might yield information from which means of reducing the rates of wear might be deduced. The aim was to determine the conditions of combustion which lead to carbon formation and, in particular, to the formation of exceptionally hard particles of carbon. It was found that whereas with Class B fuels the carbon particles deposited were both large and hard, those obtained with Class A were small and, in some instances, of a more graphitic nature and so tended to assist lubrication rather than to cause abrasion.

The work has included a study of the products of incomplete combustion. These products are usually only a very small proportion of the total exhaust, so they do not contribute greatly to the loss of thermal efficiency. They are important, however, because they frequently give rise to engine fouling and wear, particularly when lower grade fuels are employed. Formation of acids during the combustion of fuels containing sulphur has been investigated with the aid of a modified form of B.C.U.R.A. dewpoint meter. More recently, investigation has concentrated on carbon formation. In addition to

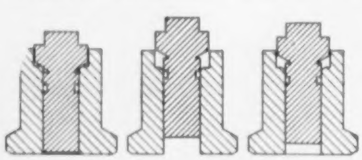


Fig. 2. From left to right, the Bicera delivery valve seated, about to unload, and during the return to its seat

chemical analysis of the deposits, comparisons have been made between the rates of carbon formation with various fuels as well as between the hardnesses of the particles. Another investigation that has been made by these laboratories is that concerning the corrosion of injection nozzles. This was described in detail in the May 1953 issue of *Automobile Engineer*.

Attention has been devoted both to the fundamental and to the more practical aspects of the combustion problem. Work on the fundamental aspect has included the use of a

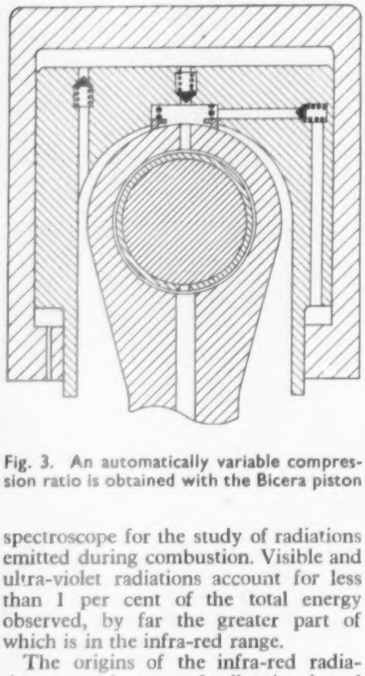


Fig. 3. An automatically variable compression ratio is obtained with the Bicera piston

spectroscope for the study of radiations emitted during combustion. Visible and ultra-violet radiations account for less than 1 per cent of the total energy observed, by far the greater part of which is in the infra-red range.

The origins of the infra-red radiations are changes of vibrational and rotational energy of the molecules and continuous radiation from solid particles. Therefore, study of infra-red spectra will reveal the type of molecules responsible for the radiation. Quantitative measurement of the radiation will indicate the temperature of the emitting body so that an estimation can be made of the radiation loss. From the thermodynamic point of view, the infra-red emissions are more important than the visible or ultra-violet, because they are closely associated with the heat release.

However, the visible and ultra-violet spectra are interesting from the point of view of chemical kinetics, because they are a means of identifying the radicals and atoms, and therefore the intermediate products before the final molecules are formed. Although the investigation has not so far been extended to cover this range, the Association hopes to be able to do so later. For the study of the infra-red range of the flame spectra, radiations transmitted through a sapphire window in the combustion chamber are directed through the stroboscope to a spectrometer and thermopile, Fig. 1. A specially designed, high-speed stroboscope geared to the engine is employed to enable readings to be made over the same two degrees of crank angle during successive firing strokes; any period of two degrees in the combustion cycle can be selected.

Bicera piston

The Bicera piston, Fig. 3, was designed to give a variable compression ratio. It was originally developed for highly turbo - charged compression ignition engines; but it has also been established that an appreciable reduction in brake specific fuel consumption at part throttle can be obtained by its application to petrol engines. This feature would make it particularly attractive for use in commercial vehicles. The best performance has been obtained at an engine speed of 1,600 r.p.m. At higher or lower speeds the gain in efficiency is not so great, but development work to improve the performance in this respect is approaching completion.

In petrol engines, although detonation at full load limits the permissible compression ratio, much higher compression ratios can be used at part load without detonation occurring. The Bicera variable ratio piston provides a progressive increase of compression ratio as the throttle is closed and therefore, by virtue of the increased expansion ratio, gives correspondingly higher thermal efficiency.

The piston comprises two main pieces: one, the inner portion or carrier, resembles a conventional piston but it is contained in the outer portion, or shell. This shell may be slid up or down relative to the inner portion to give the required variation in compression ratio. Between the top of the carrier and the underside of the crown of the shell is a chamber which is full of oil. The outer periphery of the lower end of the inner portion is shouldered and spigots into a ring fitted in the inner periphery of the base of the skirt of the shell, so that another chamber,

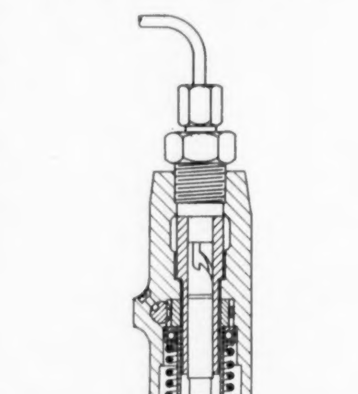


Fig. 4. Hydraulic actuation of the injection pump obviates many difficulties associated with conventional drives of mechanical pumps

also filled with oil, is formed between the shoulder and the ring.

Both the upper and lower chambers are fed with oil from the engine lubrication system. The lubricant is supplied from the big end, through a duct drilled axially in the connecting rod into an annular groove round the small end bearing housing. It then passes through a spring-loaded saddle-piece that seats on the top end of the rod and thence through drillings in the carrier into the two chambers. Non-return valves are incorporated in the passages in the carrier. Bleed holes allow a certain amount of leakage from the lower chamber to the sump, so the piston is in some degree oil-cooled.

The operation of the unit is as follows. When the gas pressure during the combustion stroke exceeds a certain value, it pushes the shell down and an oil pressure relief valve, in a vertical passage drilled through the carrier, lifts to allow some of the oil to pass out of the upper chamber to the sump, and at the same time, more oil passes into the lower chamber. The movement of the shell relative to the carrier during each stroke is small, and thus the compression ratio is changed progressively during a number of successive strokes until a balance is obtained between the combustion pressure and the blow-off pressure of the relief valve. If the engine throttle is closed slightly and the combustion pressure falls, the inertia of the shell and of the oil in the reciprocating components tends to cause the shell to return to a position giving a higher compression ratio. This movement again progresses by stages over a number of successive strokes.

During the return process, the oil in the lower chamber flows out through the bleed holes, and that in the upper one is replenished by the engine lubrication system.

At first sight, it might appear that heat dissipation difficulties would be experienced with this piston and that the valves might become sludged up and cease to function properly. However, no trouble from either of these causes has been experienced and, although a certain amount of development work is necessary to perfect the design for each application, once the correct valve setting and passage diameters have been determined, the piston appears to function correctly indefinitely.

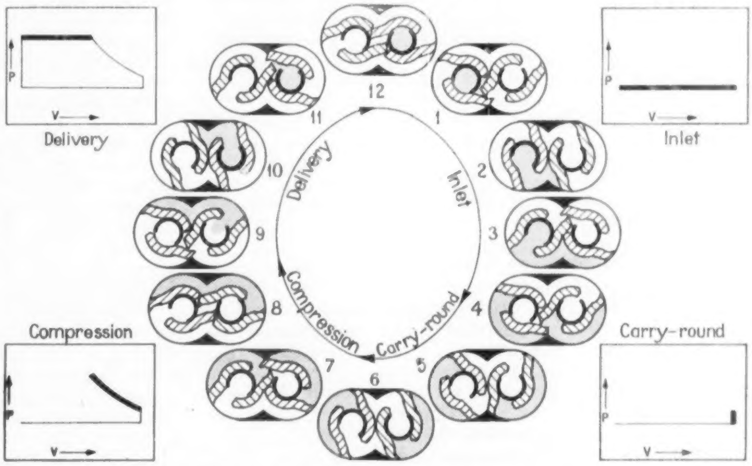


Fig. 6. The cycle of operations of the Bicera compressor

Fuel injection

Secondary injection, which is a defect commonly experienced during engine development work, is becoming increasingly troublesome as higher injection rates are now being used for turbo-charged engines. As a result of investigations on this subject, an extremely simple and effective method of eliminating the trouble has been developed.

From Fig. 2, it can be seen that the device is simply a modification of a conventional delivery valve. The modification is the counterboring of the end of the valve body to form a dash pot which surrounds the head of the delivery valve. As the valve opens, the head clears the dash pot just before the unloading portion leaves the bore of the valve body. Thus the path for the subsequent discharge is left free. When the pump spills, the unloading portion of the valve re-enters the bore and, immediately after, the valve head enters the dash pot. Then, the return motion progresses with diminishing velocity until the valve is seated. In this way, pressure waves in the delivery pipe, caused in more conventional systems by the sudden arrest of the valve as it comes against the seat, are not produced.

It has also been found that this valve may be used to induce pilot injection. Although it is well known that pilot injection reduces combustion noise and vibration, hitherto it has not been widely adopted because the methods employed to produce it have been somewhat complicated and expensive. It is difficult to conceive a simpler arrangement than the Bicera valve for effecting pilot injection. A delivery valve with a head that is a close fit in the dash pot is employed and it acts as a plunger during the initial part of the opening movement and so produces the required pilot injection. As the valve head clears the dash pot, fuel from the pipe flows back into the depression in the space it has vacated. This reverse flow causes a negative wave to pass along the pipe and seat the nozzle needle. Subsequently, injection proceeds in the normal manner.

The mechanical method of operating fuel injection pumps has limitations when high rates of injection and large delivery quantities in relation to the size of engine are required, that is, in high-pressure, turbo-charged units. These limitations arise because of the sudden and repeated application to the driving gear of the forces of several tons required to actuate the pump plungers. Obviously, the ideal to be

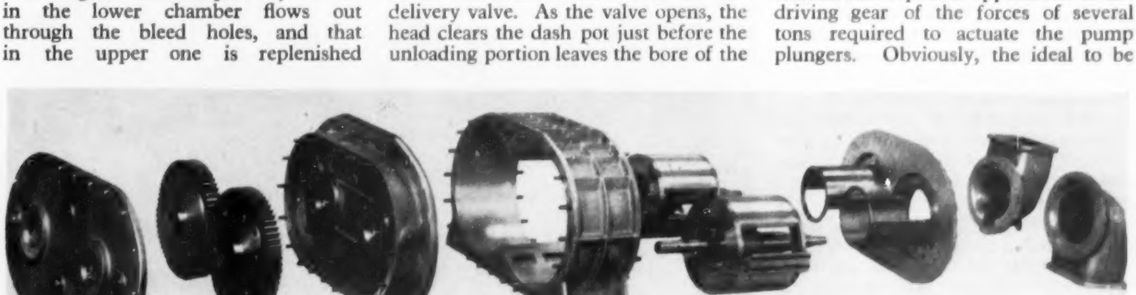
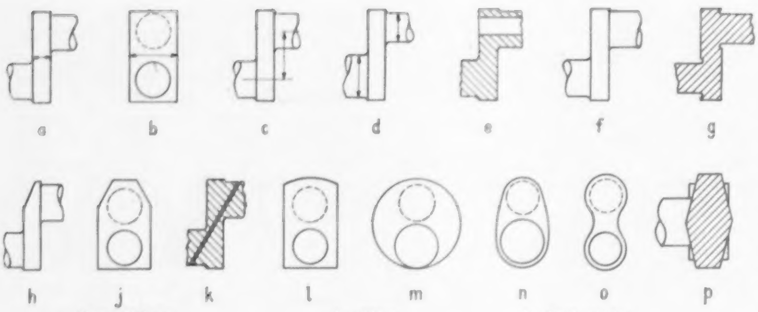


Fig. 5. The components of the Bicera rotary compressor which, unlike the Roots type blower, compresses the charge before delivery



a Web thickness
b Web width
c Length of throw
d Unequal pin and journal size
e Bored web
f Fillets
g Undercut fillets
h Back chamfer
j Side chamfer
k Oil passages
l Radiused top
m Circular web
n Oval web
o Bi-concave web
p Tapered web section

Fig. 8. The effect on crankshaft stiffness of any of these features can be evaluated by the Bicera method

aimed at is to take energy continuously from the engine crankshaft and to store it temporarily until it is required to operate the plungers.

A system based on this principle has been developed in the laboratories, Fig. 4. In this arrangement, the energy is repeatedly stored up in a hydraulic accumulator and released through a slide valve to actuate a hydraulic piston, which in turn operates the injector plunger. Fuel oil is used as the hydraulic fluid and a gear type pump driven from the crankshaft supplies it at a pressure of about 1,000 lb/in^2 to the accumulator in the base of the pump.

The advantages of this system are:

1. The substitution of a very light camshaft and drive for operating the light, balanced slide valves, in place of the heavy camshaft and drive required for conventional injection pumps.
2. A small and simple variable injection timing device may be incorporated in the camshaft drive.
3. Greater freedom in the choice of pump position, by virtue of the light drive required.
4. The optimum injection pipe pressure can be substantially maintained over the running speed range.
5. During development work, the injection rate can be altered readily by varying the servo-fluid pressure or by changing the injection nozzles.
6. The effective stroke of the plunger is a greater proportion of the full stroke than when a mechanical drive, in which nearly two-thirds of the stroke are needed for acceleration and deceleration, is employed.
7. A reduction in size can be effected by taking advantage of the ability to use a stroke the length of which is not limited by considerations of cam design.
8. Less noise.

In the pump illustrated, a Neoprene diaphragm in the base of the unit separates the oil from the air. The oil is above the diaphragm and the air below. No trouble has been experienced through deterioration of this diaphragm which, after 1,000 hours of service under full load conditions, has been found to show no signs of wear or failure. This is because the pressure exerted by the air and the oil on each side of the diaphragm are balanced. If the pressure of the oil is released, air pressure pushes the diaphragm up against a mushroom type valve which covers the outlet port for the hydraulic fluid. This prevents damage to the diaphragm which otherwise would be forced into the port. During operation, the oil is passed up to an inlet at one end of the slide valve housing. The outlet, or exhaust, port is at the other end, and between the two is the delivery port serving the piston that actuates the plunger. Movement of the valve in one direction interconnects the inlet and delivery ports and isolates the exhaust port, while movement in the other direction isolates the pressure port and interconnects the centre one with the exhaust port.

During the injection stroke, oil passes through the delivery port into the space below the piston which is lifted against the return spring and so actuates the plunger. Oil trapped between the flange round the base of the piston and a shoulder in the bore of its housing gives a dash pot effect. The trapped oil is released through a vertical groove in the bore and thence through a slot in the skirt of the piston. This slot is cut at an angle so that the timing of the point at which it passes clear of the upper end of the vertical groove bears a constant relationship to the spill timing. The pump illustrated was used for development work and is larger and more complex than would normally be necessary. Therefore, a smaller and simpler unit has since been designed for commercial application. In the illustration, a vertical rod is shown screwed in the centre of the piston; this is part of the piston-position-indicator system used in the

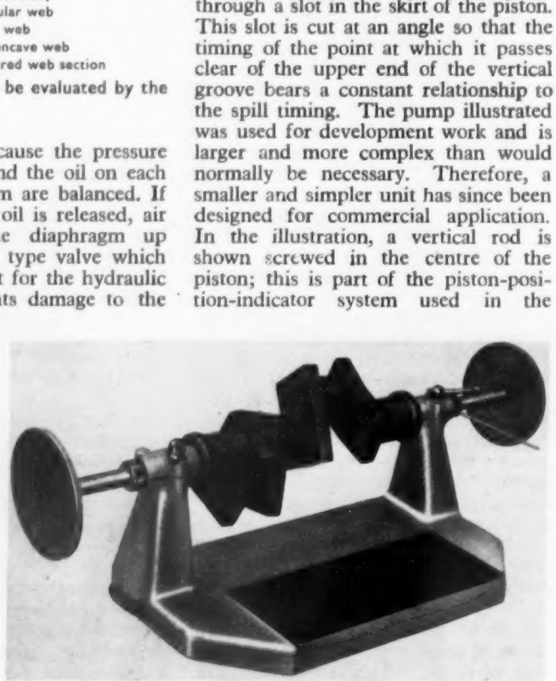


Fig. 9. From this rubber model under torsion, the nature of the displacement of the centre bearing can be seen clearly

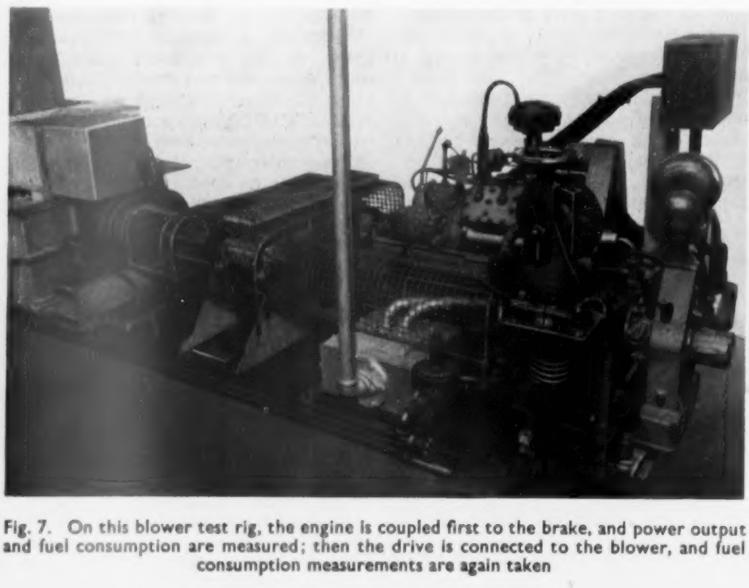


Fig. 7. On this blower test rig, the engine is coupled first to the brake, and power output and fuel consumption are measured; then the drive is connected to the blower, and fuel consumption measurements are again taken

experimental work and is not incorporated in the commercial version.

This pump, used in conjunction with the Bicera delivery valve, should be particularly attractive for applications in which quietness of operation is of prime importance. It contains more components than conventional mechanically operated units, but these are all small and less subject to wear. Because the plunger velocity in the Bicera pump is almost independent of engine speed, the valve for pilot injection is satisfactory over the whole engine speed range, whereas with a mechanically operated pump, it is effective only over a limited range.

The Bicera compressor

From the illustration showing the cycle of operations of the Bicera compressor, Fig. 6 it can be seen that the charge is compressed before the delivery port opens. Therefore, the machine works on the true compressor cycle and in this respect is superior to the Roots type of unit which simply carries volumes of fresh charge through to the delivery port, where it is compressed by the back flow. The Bicera compressor can also be used to give much higher delivery pressures than can be obtained with the Roots type. The amount of internal compression can be regulated by altering the timing of the opening of the delivery port, and the volume delivered can be varied by altering the timing of the opening of the inlet port. In this way, one compressor can be made common to a number of applications or, by interconnecting the port control and the engine control, the compressor can be made to deliver only the quantity of air required by the engine, so that the overall efficiency at light load is increased.

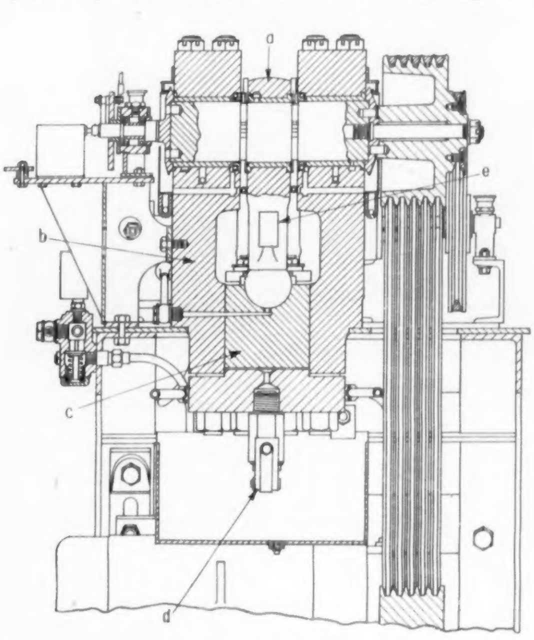


Fig. 11. A hydraulic ram loads the bearing in this test machine

Although most of the running clearances are as large as those in the Roots type blower, the internal leakage is less. This is because the charge leaking past the lobes tends to be trapped in the carry-round volume, and it is delivered during the following cycle. There is one clearance that can be smaller than in a Roots type unit and that is the one between the two rotor drums. In this blower, it does not matter whether or not the rotors touch each other, because their relative motion is one of pure rolling. As a result, the mechanical reliability of the unit is almost independent of gear wear.

All the major components of the blower are shown in Fig. 5. The rotor shafts are carried either in bearings at both ends or, alternatively, the shafts can be overhung from bearings in the drive casing. This overhung arrangement gives a more compact lubrication system and greater freedom of choice of layout of the air ducts. A rig for testing blowers is illustrated in Fig. 7.

Fig. 12. Two types of ring are employed to carry the mirrors in the torsional test rig

Crankshaft torsional vibration

Hitherto, a number of different methods have been employed for estimating the torsional stiffness of engine crankshafts. Some give reasonably accurate results for one type of shaft and others give good results for different types. However, with none can accurate estimates be made of the stiffness of every shape and size of crankshaft. This led Bicera to undertake the development of a new method suitable for general application.

The problem was to evaluate correctly the effect of every individual feature of the shaft as a contributory element of its total stiffness. With this end in view, the torsion test rigs illustrated in Figs. 10 and 13 were employed, and rubber models, Fig. 9, were used to indicate the nature of the deflections likely to be experienced. On the test rigs, the effects of fillet radii, journal and pin diameters, length of throw, bored pins and journals, and various web forms were determined, Fig. 8. One rig, Fig. 13, for test pieces comprising one throw only of different shafts, consists of a simple arm on the end of which weights can be placed to apply torsion to the test piece. An outrigger, extending to one side of the main arm, carries subsidiary weights to balance the couple due to the offset of the test piece. The weights used to apply torsion are suspended with their centres of gravity in line with the plane of reference so that they do not cause any bending. The other test rig, Fig. 10, is much larger and is designed to carry complete crankshafts and larger test pieces. Torque is applied by means

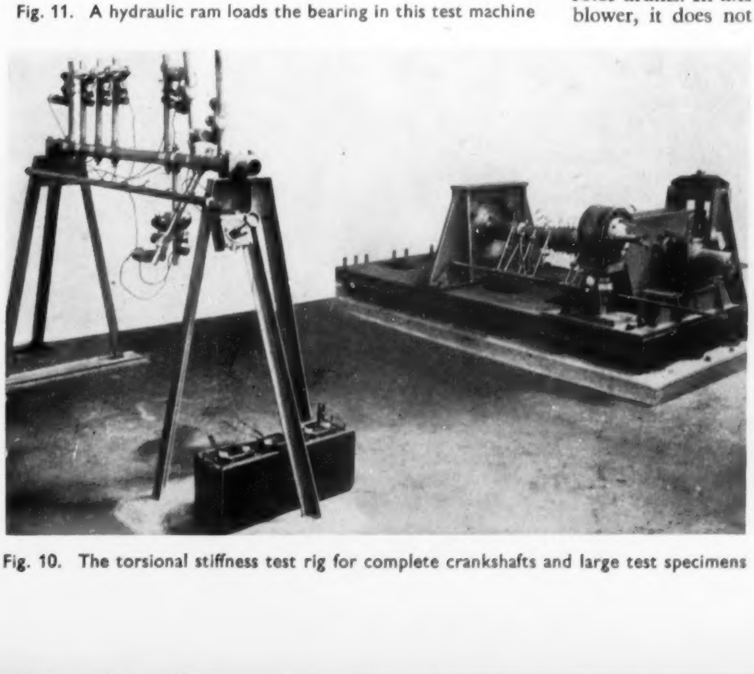


Fig. 10. The torsional stiffness test rig for complete crankshafts and large test specimens

of screw jacks acting on arms on each side, at one end of the shaft.

The optical systems employed for measuring deflections can be seen in the illustrations. A beam of light is directed through a lens, on which are cross wires, and through a collimating lens on to a mirror on the test specimen. The image of the cross wires is observed through an eye-piece, in which is incorporated a grid, mounted near the source of light. Two types of mounting, Fig. 12, can be used to carry the mirrors. One has three screws positioned radially in it and spaced 120 deg apart. These screws have needle ends which are tightened against the shaft to locate the ring accurately in the reference plane. This type of ring has the advantage that it can be easily adjusted to suit a large range of shaft diameters, but it is not so rigid as the other type, which is of thicker section and has a knife edge round its inner periphery. The ring incorporating the knife edge is divided diametrically and can be clamped on to the shaft by means of bolts in lugs at the abutting faces of the halves.

As a result of this work, it is now possible to determine the equivalent length of every individual feature affecting the flexibility of a crankshaft. The formula developed is in such a form that each factor may be taken into account separately so that it is easy during the design stage to determine directly the effect of any modification considered. This can be done simply by changing the figures in the appropriate part of the formula. Since the completion of this work, the new method has been applied to a large range of crankshaft designs, and the maximum errors encountered have been within about +3 to −6 per cent.

Because of the importance of the effect of engine damping in determining the amplitude of torsional vibrations produced at various frequencies, it was decided to carry out a thorough investigation on this subject. The work undertaken is still in progress and the experimental methods adopted are of interest. So far, a study has been made of the damping produced by the vibrational movement of the piston rings in the cylinder bore. When a crankshaft vibrates torsionally, it causes the connecting rod and piston to perform a forced oscillatory motion at the same frequency; in a medium sized engine, the amplitude of this motion may be as much as 0·020 - 0·040 in at certain crank positions.

To study this, an ingenious test rig has been devised, Fig. 14. As can be seen from the illustration, it consists of two electric motors each driving a crank and connecting rod. One motor and crank assembly causes a horizontally positioned sleeve to reciprocate inside a cylinder and the other causes a piston inside the sleeve to reciprocate at high frequencies and with much smaller amplitudes. The motion of the sleeve gives the effect of the relative motion between a piston and a cylinder in an engine, while the other crank gives rise to the same effect on the piston as torsional vibrations of the crankshaft. There are two compression rings in the piston and a connection through the sleeve is incorporated so that compressed air may be supplied to the annular space between the rings to simulate the effect of the varying gas pressure that is normally experienced in an engine cylinder. Work has also been done with this rig to determine the influence of different lubricants upon the damping obtained. Later it is hoped to evaluate the damping effect of different types of ring.

The motor producing the high

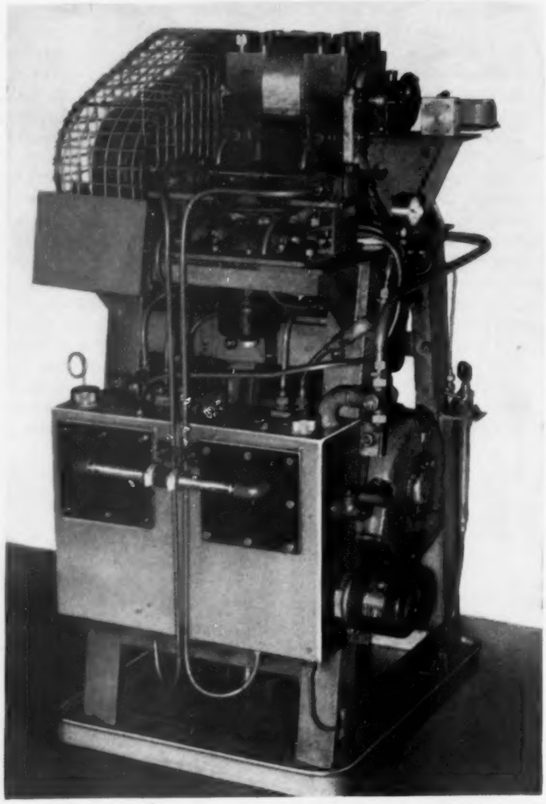


Fig. 16. In the bearing test machine, the load applied to the bearing closely resembles that obtained in an engine

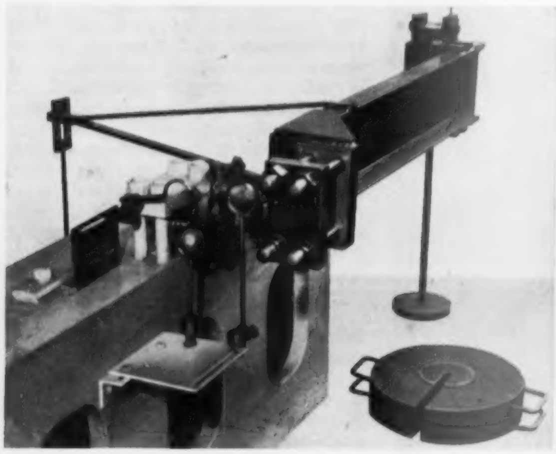


Fig. 13. This rig is employed for testing the torsional stiffness of small sections of crankshafts

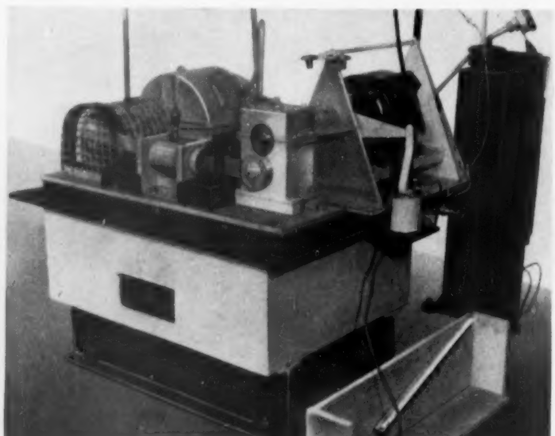


Fig. 14. The rig for investigating the damping of crankshaft vibrations by piston, ring and connecting rod assemblies

frequency vibrations is shown on the right in the illustration. It is pivot mounted so that its torque reaction may be determined. This reaction is measured by a self-centring, hydraulic device, the pressure in which is indicated by means of a sloping-tube manometer. The torque reaction is, of course, a measure of the work done to overcome the damping forces.

A large number of different types of torsional vibrational dampers for crankshafts has been studied, and the conclusion reached was that the viscous fluid type combines the advantages of simple design with considerable effectiveness in reducing amplitudes of vibration. This type of damper was originally introduced by Lanchester many years ago. It has a free mass, in the form of a ring, mounted inside a casing which is attached to the crankshaft. In the Lanchester damper, the space between the ring and its casing was filled with lubricating oil, but because of the variation of viscosity with temperature of this type of oil, the device was only partially effective. The modern version is filled with silicone fluid, the viscosity of which remains relatively constant, and this has made the design more successful.

Despite its simplicity, the viscous fluid damper is only fully effective if it is designed specifically for each application. The method now commonly employed for calculating the characteristics of this type of damper originated in America. However, experience has shown that the calculated results do not agree with those obtained in practice.

The investigations confirmed the fact that maximum power dissipation does not occur under the conditions predicted by the commonly used method of calculation. In fact, the error can be as much as 200 per cent. As a result, dampers designed in accordance with the results obtained by this formula are, in many cases, unnecessarily large.

A new method has been developed to enable the designer to determine the correct proportions of the inner mass and casing, and the detuning effect. Accurate evaluation can also be made of the vibratory torque of the engine with and without the damper fitted. Moreover, the new method also defines the maximum permissible power dissipation of the unit. It thus makes possible the use of the smallest size of damper compatible with the requirements.

Bearings and miscellaneous instrumentation

The subject of bearing design is one on which there is considerable scope for research. One of the difficulties to be overcome when undertaking such work is the provision of a means of testing bearings under conditions which are similar to those obtained in an engine, but in which the different variables such as load, lubricant supply and oil temperature can be readily controlled.

The test rig illustrated in Figs. 11 and 16 was designed to meet these requirements. From Fig. 11, it can be seen that the bearing under test is mounted in the connecting rod, a, and runs on a 3 in diameter shaft, the ends of which are supported in two more bearings. The main casting, b, of the unit incorporates a hydraulic cylinder in which is a ram, c. The centre portion of the shaft, on which the connecting rod is mounted, is slightly eccentric. This imparts a reciprocating motion to the rod, the lower end of which is spherical and bears in the ram, which therefore reciprocates with it. The cylinder is kept primed with oil at low pressure through a non-return valve. On the downward stroke of the ram, pressure builds up until the relief valve, d, opens. By adjustment of the valve, the peak pressure on the bearing can be varied from zero to 10,000 lb/in^2. A strain gauge, e, on the rod is used in conjunction with an oscillograph to indicate the bearing load.

Fig. 15. An arrangement for continuous measurement of piston temperatures

Temperature, pressure and quantity of the lubricating oil flowing through the bearing are all controllable. By fitting alternative pulleys, the shaft speed can also be changed. Thermocouples are employed to measure the bearing temperature. The loading obtained on this rig closely resembles that which would be applied to the bearing in an engine. Although such a test does not permit an accurate assessment to be made of the performance under normal running conditions, it is useful to indicate the relative merits of different bearings.

The measurement of piston temperature is an important part of engine development work and some methods of doing this were described in the June 1953 issue of *Automobile Engineer*. Another method has been developed by the staff of the BICERA laboratories. As can be seen from Fig. 15, a Tufnol rod with side plates riveted on to form a channel section houses the leads to the thermocouples, and is secured by means of three set screws to the piston skirt. The upper end of the rod is profiled to fit snugly against the skirt.

Flat strips of beryllium-copper separated by insulating strips form the leads. Adjacent to the thermocouple end they are bound securely to the Tufnol rod. They extend downwards for a few inches into the guide channel and are then turned up again and passed into a second guide channel, which is parallel to the first, and which is secured to the crankcase at a point adjacent to the cylinder skirt. From the upper end of this fixed channel, the leads are taken away to the temperature indicator. This arrangement has the advantage that a continuous connection is maintained between the thermocouples and the indicator. The alternative of incorporating contacts that are made as the piston approaches the bottom of its stroke is not entirely satisfactory because, owing to uncertainty as to whether the contact is equally good each time it is made, the somewhat slow and laborious *null method* of obtaining readings must be adopted.

When measurements of fuel consumption of an engine are taken by conventional methods, it is important that the fuel level in the measuring instrument remains constant. Otherwise, air anywhere in the system between the meter and the engine, for example in the filter, expands as the fuel head is diminished. This displaces fuel that has not been metered, and results in inaccurate consumption figures being taken. The laboratories have developed a simple but sensitive balance method for metering a known weight of liquid. This is used in conjunction with a combined stop watch and revolution counter. Its accuracy is unaffected by changes in specific gravity, it can be mounted at any height without introducing parallax errors, and the fuel level does not change while the readings are being taken.

Two other instruments have been developed at the laboratories. One is the Bicera spark unit, which incorporates the Sunbury balanced-disc pressure pick-up in conjunction with the Farnborough recording unit. The Sunbury instrument, with its rapid response, is suitable for high speed work and the Farnborough recording unit produces a large scale, undistorted diagram on paper in a matter of seconds.

The second instrument is the Bicera capacitance-type pick-up. Because the Sunbury balanced-disc unit works electro-magnetically and must be large enough to contain a coil wound round a pole piece, it cannot always be sited close to the point where the pressure is to be measured. Long passages to the pick-up introduce errors, so the Bicera capacitance type unit, in which the balanced-disc forms one pole of a polarized variable condenser, was developed to meet the need for a small diameter, long-reach pick-up. This unit can withstand much higher temperatures than electro-magnetic types.

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CORRESPONDENCE

FRONT SUSPENSION

SIR,—I beg leave to question two of the statements in the article on Front Suspension by Mr. D. R. Hume in your January issue. Firstly he states that, with divided-axle front suspension, precession due to wheel tilt is of no consequence. This is contrary to some practical experience and he surely admits the weakness of his own arguments when he states that rack-and-pinion steering is not recommended for use with this type of suspension unless some damping device is incorporated.

Secondly he shows the Ford front suspension system as having a high roll centre with small wheel tilt. The accompanying sketch suggests that the roll centre of the Ford system is substantially below hub level, the actual height being affected by the relative lengths of the swinging bottom link and the vertical leg, also the initial angles of the members. If secondary effects are ignored, there is an approximate relationship between roll centre and wheel tilt. For example, with a system employing equal length parallel transverse links, it is clear that the king pin must tilt through the same angle as the chassis when the car rolls; equally clearly, the roll centre is at ground level. The same applies with the vertical pillar type of suspension as used by Lancia. Now, consider a swing axle system pivoted at hub height on the centre line of the car; when the car rolls the wheel does not tilt at all and the roll centre is at hub height.

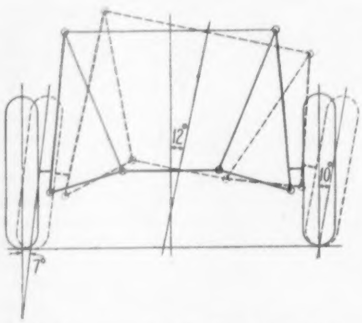
Track variation, to which Mr. Hume also refers, is taken care of very kindly by modern tyres but such variation is often considerable both with swing axle systems and some double transverse link systems. The fact that this track variation does not necessarily produce rapid tyre wear appears to be only one of the considerations involved. Another is the partial loss of adhesion, owing to the distortion of the tyre. The force involved in distorting the tyre reduces the potential force available for wheel adhesion in the case where both off and near side springs are deflected together. In the case where only one spring is deflected the tyre distortion also produces a reaction tending to deflect the car from the driver's intended line of progress.

Good handling characteristics in private cars remain a subtle and elusive quality and it seems likely that this matter of lateral tyre distortion has received insufficient attention.

R. C. SYMONDSON, A.F.C., M.A.

MR. HUME replies:—

I neither stated nor intended to give the impression that precession in a divided-axle or single-link suspension system is of no consequence. My object was to show that in a suitably designed arrangement, precession effects can be minimized to the extent that the system is acceptable. As stated in the article, by fitting light weight wheels, the effects of precession are reduced to such an extent that they are no longer noticeable if a worm-and-roller steering gear is employed. On all the divided-axle suspension systems with which I have been concerned, roller thrust bearings have been used on the swivel pins, so there has been no damping in the steering system other than that provided by the steering gear. On virtually all the current double transverse-wishbone-link arrangements, whether the



Sketch referred to by Mr. Symondson

steering is of the rack-and-pinion or any other type of gear, plain bronze or Tufnol thrust washers are used. However, double-row, roller thrust bearings are employed in the Ford system.

When rack-and-pinion steering gear is fitted, a hydraulic damper is only necessary, of course, if the wheels used are of such a weight that the effects of precession are noticeable. The height of the roll centre, in both double- and single-link arrangements as well as in the Ford system, as Mr. Symondson states, depends on the length and disposition of the links. For instance, by moving the pivots at the inner ends of the lower swinging links of the Ford system downwards until the links are horizontal when in the static position, the roll centre may be assumed to be at ground level. However, if the angle of the links to the horizontal is increased by moving these pivots upwards, the roll centre is raised. Thus, it is important to consider the detail dimensions and disposition of the links in the system, and this cannot be done with Mr. Symondson's sketch, since he states that it is out of scale.

In order that Fig. 6 of the article should fit in the space available on the page, it also was drawn out of scale. The roll centre was shown as being approximately 20 per cent of the rolling radius below the height of the wheel axes, although in actual fact it is 35 per cent below. This position of the roll centre is still relatively high as compared with double-transverse-link arrangements, in which the roll centre is 90-100 per cent of the rolling radius below the height of the wheel axes. With the single-link suspension of the Allard Palm Beach, the roll centre is 20 per cent below the wheel axes.

An important characteristic of the Ford system is that for the first 5-6 deg of roll, the roll centre remains approximately static, but any further increase causes it to become progressively lower. However, at a roll angle of 5-6 deg, with a double-transverse-link arrangement, in which the roll centre is 1 in above ground level in the static laden position, the roll centre is approximately 1 in below the ground level. It can be seen that the Ford system has an advantage, so far as roll resistance is concerned, by virtue of the small distance, and therefore moment, between the roll axis and the centre of gravity of the sprung mass. Moreover, the wheels are attached rigidly to the vertical legs and therefore the camber angle cannot change to such an extent as to encourage roll under lateral loading. This is a good characteristic also of the single-transverse-link systems although, in this case, the change of camber angle is such as to oppose the tendency to roll.

The small amount of wheel tilt experienced with the Ford suspension system, referred to in the article, was that obtained when one wheel traverses a bump with the vehicle on an even keel, although this perhaps was not made clear. Tyre distortion induced with divided-axle systems as the wheel rises and falls depends, of course, on the ratio R:L, where R is the rolling radius and L the distance between the link pivot and the centre of contact between the tyre and the road. The adoption of smaller wheels and a fairly wide track tends to reduce track variation and consequently tyre distortion.

However, the nature of the divided-axle system is such that the tyre assumes a positive angle of attack, that is, the camber angle of the wheel becomes negative. On the other hand, with double-transverse-link systems, with parallel, equal length links, the wheel is pushed in and out and there is a tendency for negative angles of attack to develop. Some unequal length double transverse link arrangements give virtually no track variation.

Winter Grip Tyre

DUNLOP's new Winter Grip tyre has been specially designed for very bad conditions of snow and ice. It has over 2,000 inches of gripping edges, spaced to give the maximum squeegee action on ice While this new tyre gives high traction and stability, it also allows normal speeds on clear roads.

PRECISION PLAIN BEARINGS

Two Units for Applications Where Extremely Accurate Radial Location of the Shaft is Essential

TWO new bearings for applications where shaft rigidity and precise radial location are of critical importance, have been introduced by the Glacier Metal Co. Ltd., Alperton, Wembley, Middlesex. One, the Micro Clearance bearing, is a plain bearing designed for use where close running clearances are necessary positively to locate the journal. An important advantage that this unit has over more conventional sleeve type bearings is that its temperature during operation remains at a relatively low level. This minimizes any tendency for thermal expansion to occur and cause the assembly, of which it is a part, to be displaced from its true position, or for failure to take place as a result of the running clearances being reduced.

This new bearing can be described as a development of the Mackenson bearing, which is also designed for fine clearance conditions. The Mackenson bearing is a sleeve type unit of conventional form but on its outer periphery there are at least three radial projections, or ribs, parallel to the central axis. When the bearing is inserted into its housing, it is located by means of set screws carried radially in the housing and tightened against the ribs. Alternatively, the housing and the ribs can be tapered, so that adjusting nuts at each end can be used to pull the bearing into the taper to apply the pressure to the ribs. In each case, this pressure causes high areas to be formed locally in the bore, Fig. 1. At these high spots, the clearance between the shaft and the bore of the bearing is small, while between them it is relatively large.

During running, oil is introduced at one or more of the areas of maximum clearance, and as it passes into the areas of minimum clearance, a hydrodynamic wedge is formed. Since the whole of the space between the shaft and the bearing is filled with lubricant, oil shear occurs over the whole surface of the bearing, that is, in both the load-carrying and the load-free regions. This causes a degree of heat generation which, although satisfactory for many normal applications, may cause thermal distortion not acceptable in mechanisms such as precision machine tools.

The Micro Clearance bearing is designed to minimize heat generation. It comprises a steel shell lined with

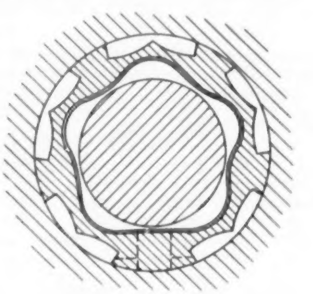


Fig. 1. In this diagram of the Micro Clearance bearing, the deformation of the bearing bore is shown exaggerated to illustrate the principle of operation

0·010-0·015 in Findlays Motor Metal L1, which is a high-quality, tin-based white metal. Five radially extending ribs are formed on its outer periphery; they are parallel over the length of the bearing. Their dimensions are closely controlled to provide an interference fit in the housing, which must also be machined accurately. When the unit is pressed into its housing, the bore is displaced inwards in the areas adjacent to the ribs.

One rib, that which is at the bottom when the bearing is installed, is wider than the others and has one or more slots machined radially through it over almost its whole length, Fig. 2. Each slot accommodates a felt pad to feed oil to the bearing surface. The oil is contained in reservoirs formed either on each side of the bottom rib in the space between the housing and the bearing, or in the housing beneath the bottom rib. If the reservoir is in the space between the bearing and the bore of its housing, one end may be sealed by a screwed-on plate, and the other end is usually shouldered. Tangential slots in the rib feed the oil into the longitudinal slot containing the felt pad.

This bearing has two main advantages over the Mackenson type. One is that whereas the Mackenson unit is generally made of bronze, which has a relatively high coefficient of expansion and therefore, as the temperature rises, tends to expand in the direction of the shaft, the steel shell of the Micro Clearance bearing restricts inward expansion so there is less tendency for the running clearance to close and cause shaft seizure. The other advantage is that the oil is introduced by the felt pad at a point of minimum clearance so that only a thin film is maintained on the surface of the journal. Therefore, this film only contacts the housing at the points of minimum clearance, which are not large in area. As a result, the temperature rise due to oil shear is relatively small.

Fifteen standard sizes of Glacier Micro Clearance bearings are available from stock for shafts ranging from 1 in to 4½ in diameter. The tolerances on the outside diameter of the bearing and the bore of the housing are, of course, critical. They vary with the size of the bearing, from 0-0·0008 in to 0-0·0016 in. Shafts must be ground accurately to suit each individual assembly and a 0·0003-0·0006 in running clearance must be allowed. Each bearing is labelled by the Company's inspection department to show its minimum wall thickness.

The other new design is called the Film Located bearing. It has a higher load carrying capacity than the Micro Clearance bearing and runs at a temperature which is maintained constantly at a relatively low value, but which is nevertheless higher than that generated in the Micro Clearance bearing. A Film Located bearing comprises a steel shell lined with Findlays Motor Metal L1. Thirteen sizes, suitable for shafts of between 2 in and 5 in nominal diameters, are available.

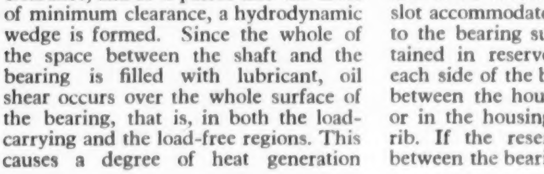


Fig. 2. The slots in the base of the Micro Clearance bearing are incorporated to accommodate felt pads that spread a thin lubrication film over the journal surface

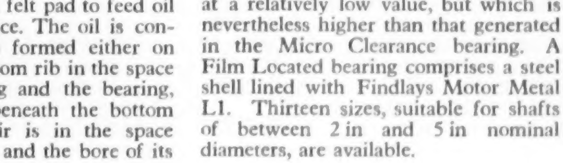


Fig. 3. Oil is introduced at the point of maximum clearance in the Film Located bearing, which has a spiral profile and which is positively located in its housing by a dowel

In this type of bearing the bore, instead of being truly circular, has a spiral profile, see Fig. 3, with a fairly short ramp between the points of maximum and minimum radius. The oil is introduced at the point of maximum clearance. From the illustration it can be seen that, with the shaft centrally positioned and revolving clockwise, lubricant dragged round with it is forced into a gradually decreasing clearance. The pressure developed in the oil film, as a result of this hydrodynamic wedge effect, is approximately proportional to the ratio of profile slope to radial clearance.

If a heavy load is applied in the direction of the arrow "a" and a minute shaft displacement occurs, this displacement is corrected by an increase of oil pressure in the area under load and a decrease of pressure in the area diametrically opposite it. The increase and decrease of pressure take place because of the increase and decrease respectively of the profile slope to clearance ratios in those areas. It is said that this wedge type oil film supports the shaft very firmly, and can resist in any direction considerable loads tending to displace the shaft from its true position when it is rotating in the bore of the bearing.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

The Motor Vehicle

By K. Newton, M.C., B.Sc., A.C.G.I., A.M.Inst.C.E., M.I.Mech.E., and W. Steeds, O.B.E., B.Sc., A.C.G.I., M.I.Mech.E.

London: Iliffe and Sons Ltd., Dorset House, Stamford Street, London, S.E.1. 1953. 5½ × 8¾. 590 pp. Price 35s.

Since its original publication in 1929, this book has become well established as a standard work. In the preface to the recently introduced fifth edition, it is stated that the necessity for reprinting to meet the sustained demand has afforded an opportunity to remove one or two minor ambiguities that had become apparent, to bring up-to-date certain points in the chapter on fuels, and to introduce additional matter in connection with the S.U. carburettor; but it appears that major revision has not been necessary.

As in earlier editions, Part I deals with fundamentals of mechanics and begins with a chapter that explains the principles of engineering drawing. Part II is devoted to a study of automobile engines of both petrol and diesel types, and attention is also given to such matters as fuels and lubrication. The third part deals with gearbox and other transmission components, axles, brakes, steering, chassis details and suspension.

This book contains more than 500 illustrations prepared from drawings and photographs; these provide a valuable supplement to the text. Non-technical language is used so far as possible and mathematics have been largely excluded. The aim of the authors has been to provide an accurate but not unduly technical explanation of automobile engineering theory, and to keep the whole work in a logical sequence.

Lexique Illustré de l'Automobile

Paris: S.N.E.E.P., Bibliothèque de l'Argus, 1, Place Boïeldieu, Paris. 1953. 21 × 27 cm. 220 pp. Price Fr 1,600 with paper cover and Fr 1,950 with stiff cover.

The fourth edition of this book is in general similar to the previous edition, except that it has been checked by additional authorities in different countries with a view to ensuring that the most commonly used technical terms for each component are given. Perfection in this respect has not yet been attained, and, considering the disagreement that exists between various authorities, it is doubtful whether even near perfection could ever be achieved.

Although there are a number of technical dictionaries available, some in two languages and others in more than two, these generally do not satisfy the requirements of automobile engineers, in that they do not give the translations of many of the technical terms peculiar to the motor industry. The Lexique Illustré de l'Automobile gives the technical terms in French, English, German, Italian and Spanish for nearly all the components used in the construction of cycles and motor vehicles, including motor cycles. There can be little doubt that everyone whose business it is to study foreign technical literature concerning these industries, or who is interested in such works, should have available a copy of this book.

After the title pages, there follow five contents pages, one for each language, in which the major assemblies and the pages in which they are dealt with are listed. After this come 170 pages in which on each left-hand page there is a diagram showing exploded views of the major assemblies, the components of which are numbered for reference purposes, and listed in five columns on the right-hand page. Each of the five columns is, of course, for a different language. At the end of the book are four pages on which are conversion tables giving metric, English and American equivalents of the different units used in engineering. These tables are followed by five alphabetical indexes, one for each language, of components.

One of the most valuable features of this book is the illustrations, since they help to avoid ambiguity and doubt when determining the correct translation for terms that can be used for more than one component. The field covered is extensive. For instance, in the transmission section, not only are conventional single dry plate clutches and fluid flywheel components listed, but also automatic transmission parts and several synchromesh units are included. A number of different front suspension systems, including front wheel drive arrangements, are dealt with, but inevitably there are some omissions, for instance, a list of the different types of suspension layouts would be valuable. However, the most useful feature of this work is its specialist approach, which makes it one of the best of its kind in the world.

The Autocar Handbook

By the Staff of "The Autocar."

London: Iliffe and Sons Ltd., Dorset House, Stamford Street, London, S.E.1. 1953. 5½ × 8¾. 216 pp. Price 7s. 6d.

The Autocar Handbook has been written especially for the motorist who, without going too deeply into engineering details, wishes to have an understanding of his vehicle to help him to obtain the best possible service from it on the road. As such, it should be of interest to the owners both of current models and of older vehicles. In addition, it may prove to be a useful work of reference to automobile engineers who from time to time wish to refresh their memories concerning elementary principles.

For more than 45 years, successive editions of this book have reflected the many developments in automobile engineering, and kept readers up-to-date. In this 21st edition, a new approach has been made to the subject, and the work has been entirely rewritten. After a brief description of the fundamental principles involved, practical features of current design and the reasons for them are given. It is not possible, of course, in a work of this kind to treat each individual component in great detail, but every important part of the car's structure and mechanism is considered. A noteworthy feature of the work is the wealth of illustrations it contains. Over two hundred drawings show clearly the construction and operation of almost every component, thus providing a complete picture of the modern car.

AN OIL BURNING HEATER

The Key-Leather K-L 10,000 Heater Unit which Burns Diesel Oil or Paraffin

UNDER certain operating conditions, even in this country, heaters in which water from the engine cooling system is used in the heat exchanger are not entirely satisfactory, because of the length of time required for the unit to come into operation after a start from cold. So far as coaches are concerned, because first impressions of the vehicle tend to have a lasting influence, it is highly desirable that the passengers should be made comfortable right from the outset of the journey. For this reason, there is a demand for oil burning heater units both here and in colder countries overseas.

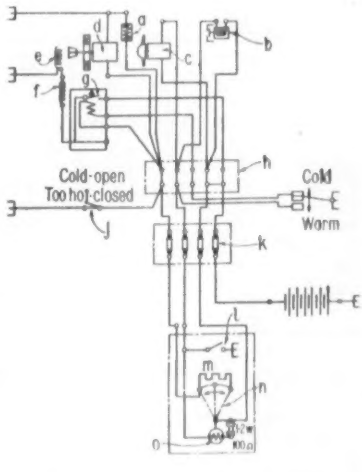
In Britain, the Key-Leather Co. Ltd., of London, E.9, have recently introduced a heater in which either diesel oil or paraffin can be burned. It is based on the Eberspacher system. The heater unit, together with the four-gallon fuel tank, is mounted on a frame, and an air intake filter can be fitted if required. Alternatively, the heater unit can be supplied alone.

A cylindrical steel casing encloses the heater unit, which is an assembly of three main components mounted one behind the other. These components are: first, the motor for the fresh air fan; second, the unit comprising the fuel pump, another motor, the burner and its blower, and third, the heat exchanger. By employing separate motors to drive the fresh air fan and the fuel pump and blower unit for the burner, it is possible to use the system to provide unheated fresh air if required. Moreover, by switching off the motor serving the burner assembly before the other one, the heater components can be cooled and the unburned gases blown away after the unit has been stopped.

This is done automatically, and obviates the danger of explosion of vapour left in the system after it has been switched off. An electrically operated valve in the air intake to the burner is open so long as the fresh air motor is running. Thus, when the burner motor is stopped, fresh air continues to be blown through the burner and heat exchanger until the current to the fresh air motor is cut off by the thermo-switch. This switch comes into operation when the temperature of the unit drops to about 40 deg C.

From the tank, fuel is supplied through a small filter unit to a union on the base of the gear-type fuel pump. This pump is driven at one end of the electric motor, the other end of which drives the blower serving the heater. From the pump, the fuel is passed through a pipe line to an electrically operated fuel valve and thence to the burner. This valve is opened when the motor is switched on and closes when it stops. Therefore, there is no possibility of fuel seepage into the unit when it is not in operation. Without the fuel valve, there would be a danger of fire on starting the unit.

Mounted at the centre of the rear face of the blower rotor is a cupped steel pressing, with a flanged rim. Fuel is fed into this cup and is sprayed off the rim by centrifugal force so that it is well atomized. The air intake for the burner is a pipe passed radially into the unit. Incoming air is directed over the electric motor to keep it cool, and is then drawn into the centre of the rotor whence it is thrown outwards and passed between guide vanes and out through an annular nozzle round the cupped pressing of the burner. This nozzle directs the air on to the fuel spray and thus helps further to atomize it.

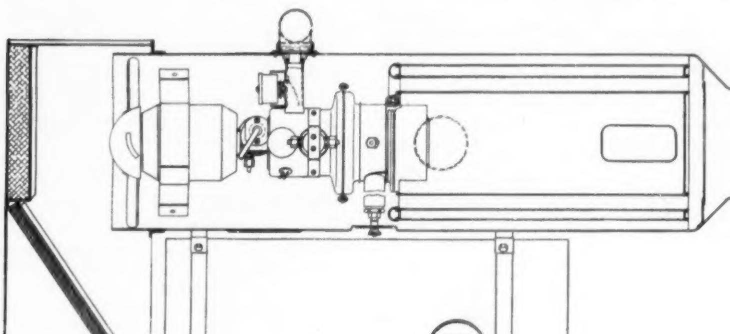


a Fuel valve
b Air valve
c Fresh air fan
d Combustion air blower
e Heater plug
f Series resistance
g Heater plug relay
h Terminal board
j Safety switch
k Fuse box
l Pull switch
m Eight ohm resistance
n Turn switch
o Pilot lamp

The electrical circuit

A heater plug for starting is fitted. The time required fully to ignite the burner is about 40 seconds; approximately 60-80 seconds later, the heater plug is automatically switched off and the fresh air fan switched on. The fresh air passes from the front of the unit round the annular space between the burner and the main outer casing, and thence through the heat exchanger unit to the outlet at the rear. With this arrangement, the incoming fresh air helps to cool the fuel supply lines and pump, and the two electric motors.

The heat exchanger comprises three concentric steel tubes. Hot gases are discharged from the burner through a hole in the circular plate that closes the front end of the centre tube. They pass along the length of this tube, the rear end of which is also closed, and then outwards through large radial ducts communicating with an annular space between two more tubes. The ends of this space are each closed by a steel ring welded between the tubes, so the hot gases, after passing forward again, go out through another radial duct into the exhaust pipe. The fresh air is heated as it passes along the spaces between the annular component of the

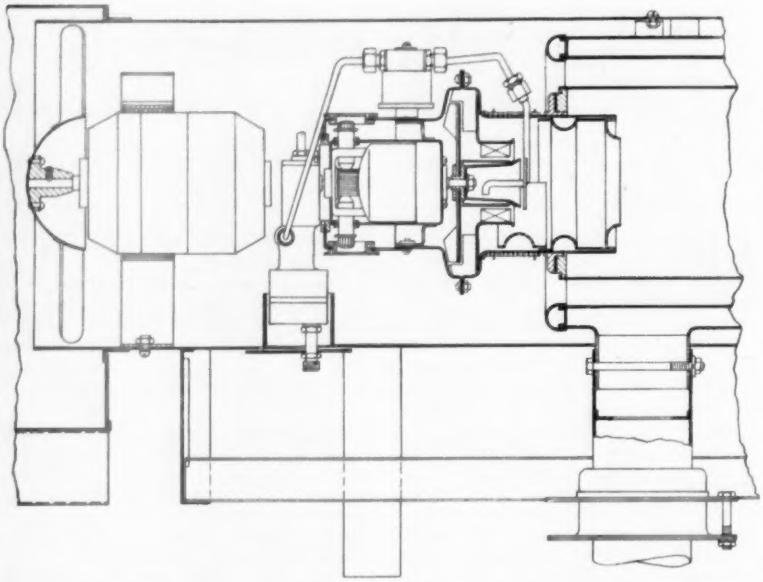


The heater unit can be supplied alone or mounted, together with a filter and fuel tank, on a frame

heat exchanger and the housing, and between the annular component and the centre tube.

A switch unit on the dash is used to control the unit. By actuating a pull switch alone, only fresh air is delivered to the interior of the vehicle. On the other hand, if the heater is to be used, a three-position switch is turned from the "off" to the "full" position. This starts the electric motor driving the blower that serves the burner, and opens the fuel valve. At the same time, the heater plug relay is energized and the plug is switched on. When the fuel is burning properly, a thermo-switch near the fresh air discharge port breaks the circuit to the heater plug and at the same time makes the circuit serving the fresh air fan and an indicator lamp on the dash. If the switch is then turned to the "half" position, the speed of the electric motor driving the fuel pump and blower for the heater is reduced.

A number of safety precautions have been incorporated in the design. Some have already been mentioned, for instance, the electric fuel cut-off valve which ensures that leakage into the burner cannot take place while the unit is not in use, the supply of cooling air over the fuel pump and electric motor, the automatic scavenging of the burner chamber when the unit is switched off, and the automatic switching off of the current to the heater plug when the unit is burning properly. In addition, there is a safety switch in the hot air outlet port. This switch cuts out the motor driving the fuel pump and blower serving the burner if the outlet temperature exceeds 150 deg C.



Separate motors are employed to drive the fresh air fan and the blower serving the burner

When the control switch is in the "full" position, the heat output of the unit is 40,000 B.T.U/hr and the fuel consumption is 2½ pt/hr; in the "half" position, the output is 24,000 B.T.U/hr and the consumption is 1·4 pt/hr. The temperature differences between the incoming and outgoing fresh air, with the switch in the "full" and "half" positions respectively, are about 82 deg C and 50 deg C. A constant flow of about 17,600 ft^3 of air is passed per hour. The maximum power consumed is 140 watts and the unit can be supplied for use with 12 or 24 volt systems.

The diameter of the heating element is 9·4 in, its overall length is 39·4 in, and the diameter of the fresh air pipes is 5·9 in. When installed, the assembly, comprising the heater, fuel tank, air filter and frame, occupies a space 43 in long by 22½ in wide by 15½ in deep. The weight is approximately 55 lb.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

BIRMINGHAM CENTRE

Tuesday, 23rd March, 6.45 p.m. General Meeting in the James Watt Memorial Institute, Great Charles Street. Paper: "The Motor Cycle as a Utility Machine," by D. W. Munro, M.I.Mech.E.

NORTH-EASTERN CENTRE

Wednesday, 17th March, 7.30 p.m. General Meeting in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Some Problems in Lubrication and these Substances Called Additives," by A. Towle, M.Sc., M.I.Mech.E.

NORTH-WESTERN CENTRE

Thursday, 25th March, 6.30 p.m. for 7 p.m. Annual Dinner in the Engineers' Club, Albert Square, Manchester. Address by the Chairman of the Automobile Division, Professor S. J. Davies, D.Sc., Ph.D., Wh.Ex., M.I.Mech.E., entitled 'Combustion in Compression-Ignition Oil Engines."

SCOTTISH CENTRE

Monday, 15th March, 7.30 p.m. General Meeting in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Operation Experiences with a 75 kW Gas Turbine," by G. B. R. Feilden, M.A., M.I.Mech.E.

WESTERN CENTRE

Thursday, 11th March, 6.45 p.m. General Meeting in Fortes Restaurant, Milsom Street, Bath. Paper: "Tyre Characteristics as Applicable to Vehicle Stability Problems," by T. J. P. Joy, B.Sc.(Eng.), A.M.I.Mech.E., and D. C. Hartley, B.A.

Thursday, 25th March, 6.45 p.m. General Meeting in the Grand Hotel, Bristol. Paper: "Problems in the Design of an Economical Automobile Transmission," by T. C. F. Stott, M.I.Mech.E.

COVENTRY GRADUATES

Wednesday, 17th March, 7.30 p.m. in the Craven Arms Hotel, High Street, Coventry. Annual General Meeting followed by a General Meeting. Paper: "Brakes," by G. J. H. Buss, B.Sc., G.I.Mech.E.

LONDON

Tuesday, 13th April, 5.30 p.m. at Storey's Gate, St. James's Park, S.W.1. Paper: "Tyre Characteristics as Applicable to Vehicle Stability Problems," by T. J. P. Joy, B.Sc.(Eng.), A.M.I.Mech.E., and D. C. Hartley, B.A.

NORTH-EASTERN CENTRE

Wednesday, 21st April, 7.30 p.m. General Meeting in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Operation Experiences with a 75 kW Gas Turbine," by G. B. R. Feilden, M.A., M.I.Mech.E.

NORTH-WESTERN CENTRE

Monday, 5th April, 7.15 p.m. General Meeting at Leyland Motors, Limited, Leyland. Paper: "Tyre Characteristics as Applicable to Vehicle Stability Problems," by T. J. P. Joy, B.Sc.(Eng.), A.M.I.Mech.E., and D. C. Hartley, B.A.

WESTERN CENTRE

Thursday, 29th April, 6.45 p.m. General Meeting in the Bristol Aeroplane Company, Car Division. Paper: "Brakes for High-speed Automobiles," by J. A. Channer, A.F.R.Ae.S.

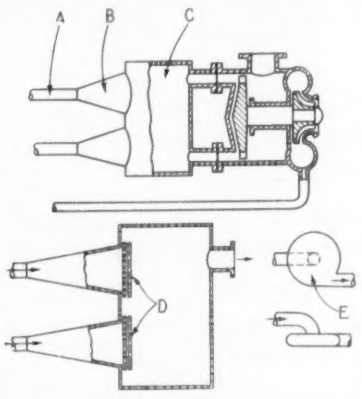
CURRENT PATENTS

A Review of Recent Automobile Specifications

Turbo-charged diesel engines

ALTHOUGH this invention would appear to be more particularly intended for engines larger than those of road transport vehicles, it is of considerable interest as it aims to realize the somewhat conflicting advantages of both "blow-down" and "steady-pressure" exhaust turbines in a single installation. In the first system, a widely fluctuating pressure in the small diameter exhaust ducts from individual cylinders secures an increase in the output of the engine that tends to obscure its ill-effect on the efficiency of the turbine.

Accordingly, the individual ducts of suitable groups of non-interfering cylinders are combined to form two exhaust ducts A which deliver the gases through diffusers B to a chamber C, whence they flow at a common pressure through two short ducts to the turbine nozzle assembly. By the conversion of velocity into pressure at the diffusers, negative pressure impulses are returned in the exhaust ducts and assist the cylinder expulsion and scavenging processes. At the same time a substantially constant pressure is maintained at the turbine nozzle.

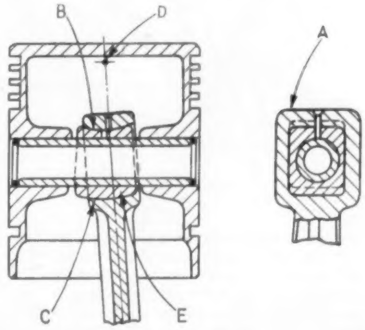


No. 700182

It may be desirable, particularly for engines operating at relatively low rotational speeds, to provide some form of non-return valve at the inlets to chamber C to prevent gases at a higher pressure in the chamber flowing back into the exhaust ducts which are subjected to waves of negative pressure. These may be plate valves, as at D, or so-called non-return throttles E in which the gases enter axially and leave tangentially. *Patent No.* 700182. *Maschinenfabrik Augsburg-Nurnberg A.G.* (*Germany*).

Connecting rod small end

THIS arrangement avoids the need for absolute perpendicularity of the gudgeon pin and connecting rod axes and allows the piston a limited freedom to align itself in the cylinder bore. These advantages, however, are only gained at the expense of a heavier construction more costly to produce than the conventional.



No. 701194

The small end A of the connecting rod is formed with an opening of rectangular section, the sides of which are parallel to a plane containing the rod axis while upper and lower arcuate surfaces B and C are struck from a common centre D on the rod axis. A bush E, freely floating on a gudgeon pin located in the piston bosses by spring clips, is of complementary shape and slidable in the eye of the connecting rod.

In an alternative construction, to facilitate machining of the rod, the bush is inverted and engaged by upper and lower arcuate-faced blocks flange located in a strictly rectangular eye in the rod. *Patent No.* 701194. *C. Bodda* (*Italy*).

Damping wheel suspensions

ROAD roar, arising from high frequency vibrations generated at the tyre on certain types of road surface, is more pronounced on vehicles having independent wheel suspension because of the substantial reduction in unsprung weight. As a corrective measure, this invention proposes to absorb the horizontal impulses by means of a connection including a rubber element pre-loaded in shear.

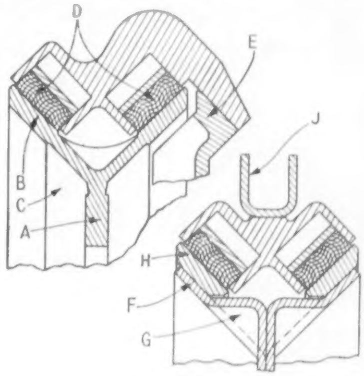
In an independent front suspension of the wishbone type, the resilient element A in the form of a rubber sandwich is secured to the lower arm B as closely adjacent as possible to the point having the greatest amplitude of movement. To the top plate of the sandwich is bolted a rod C extending horizontally forward to intersect the projected axis of oscillation of arm B. There its screw-threaded end is anchored in a pair of rubber bushes D mounted in a bracket depending from the main frame. The effective length of the rod is such that the sandwich is pre-loaded in shear, as shown, and the extent of the loading may be varied by adjustment of the anchoring nuts.

The specification also describes the application of the invention to the transverse leaf spring, quarter-elliptic leaf spring, and trailing-arm types of suspensions. *Patent No.* 700840. *Austin Motor Co., Ltd.*

Vee-type disc brake

THE rotating disc of this brake has a bifurcated rim presenting oppositely inclined, outwardly directed braking faces. Two or more pairs of friction pads, angularly spaced in either a common housing or individual housings, are applied by fluid at a common pressure from a pedal-operated master cylinder.

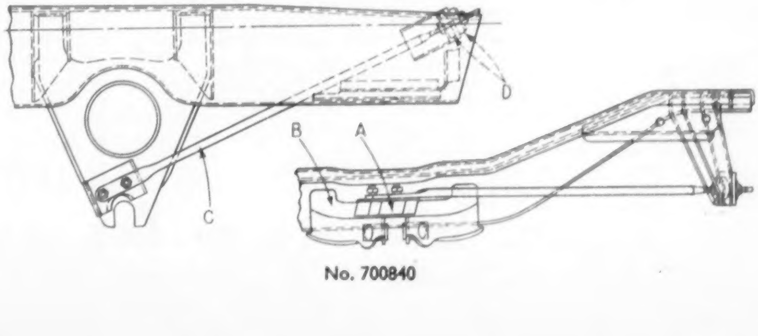
Disc A has divergent peripheral flanges B which are thickened over the operative areas and are stiffened by radial ribs C.



No. 700724

Friction pads D, each backed by a steel disc, are mounted in blind bores in a housing welded or otherwise attached to a flanged anchor plate E secured to the axle casing.

An alternative arrangement, presumably intended for a transmission brake, shows a disc formed by two complementary pressings F stiffened by pressed-out webs G. Braking surfaces are provided by frusto-conical rings H welded or copper-brazed to the flanges. In this case the housing carrying the friction pads is secured to a U-section pressing J adapted for attachment to a part of the vehicle chassis. *Patent No.* 700724. *Girling, Ltd.*

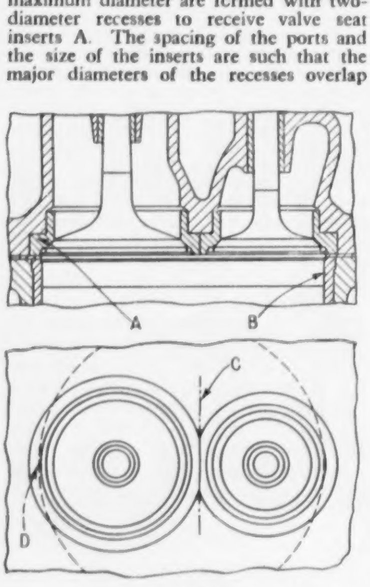


No. 700840

Valve seat inserts

IN attempts to increase the area of valve ports in O.H.V. engines the distance between the valve openings is commonly reduced and, in order to maintain wall thickness, the water space must necessarily be moved farther away from the face of the cylinder head. Particularly in diesel engines, in which the temperature of the combustion gases is high, the concentration of heat at this point at certain periods of the cycle is likely to be severe and may result in cracking or failure of the cast metal. While seat inserts of heat-resisting material can withstand the temperature, conversely they usually require the valve openings to be reduced.

These difficulties are circumvented by the invention in which valve ports of maximum diameter are formed with two-diameter recesses to receive valve seat inserts A. The spacing of the ports and the size of the inserts are such that the major diameters of the recesses overlap



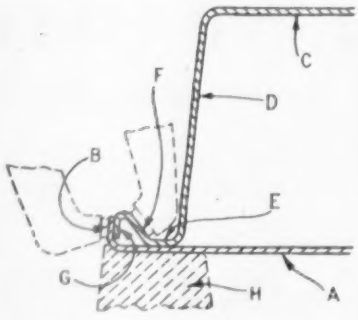
No. 700846

the bore of the cylinder or liner B and also intersect each other at a plane C. Accordingly, the inserts are formed with complementary flats which are in abutment on plane C and prevent relative rotation. On assembly the inserts, which are a press fit on their minor diameters and have a slight clearance on their major diameters, are ground on their lower faces flush with the face of the cylinder head.

Although the face of the water space between the ports is raised it will be noted that the cast metal is at this point shielded from the hot combustion gas flame by the abutting portions of the inserts. When the cylinder head is tightened on the cylinder block against the usual gasket the inserts are securely clamped by the overlaps at D. *Patent No.* 700846. *Hercules Motor Corporation* (*U.S.A.*).

Welding door panels

BY this construction inner and outer panels of a door are welded at their marginal flanges so that no weld spots are visible when the door is within its opening. Outer panel A has an inwardly directed flange B standing from the margin constituting the overlap. Inner panel C has a wide outwardly directed flange D, forming the main frame, and an overlap E which is continued by an outwardly open V-section channelled



No. 701421

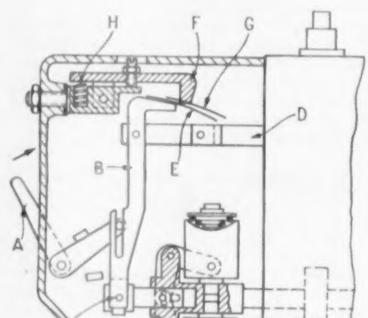
margin comprising legs F and G. With the panels assembled in position and resting on a support H, flange B is in overlapping engagement with leg G, and the electrodes of a pincer-type spot welder are applied respectively to flange B and leg F. Welding pressure and current are transmitted over leg F to leg G. Jagged edges are avoided since flange B is lower than the apex of legs F and G.

The method is applicable to other structures such as boot lids and hollow beams. In a modification, holes may be provided in leg F to permit the welding electrode to engage directly leg G. *Patent No.* 701421. *Budd Company* (*U.S.A.*).

Injection pump starting control

TO facilitate the starting of diesel engines it is common practice to fit an overriding control by means of which the delivery of fuel by the injection pump may temporarily exceed the normal limited full-load quantity. The object of the invention is to provide a control that can be operated only when the engine has been stopped. In the illustration the parts of the control system on the injection pump are shown in position prior to starting up.

The driver moves the control lever A in the direction indicated and, by means of a pin and slot connection, intermediate lever B, pivoted at C on the governor sleeve, moves the control rack D to a position beyond that for full-load delivery. During this operation the head of lever B moves along arc E clear of the full-load stop F but, when the engine starts, the governor rocks lever B about the pin of

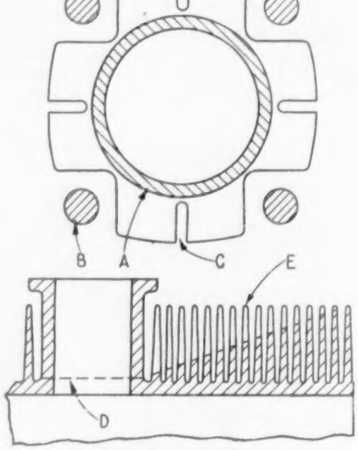


No. 701662

control lever A and the head moves along arc G. Thus the pivoted stop F is lifted against the constraint of its spring H. Immediately the driver returns the control lever to either the no-load or an operational position, stop F is restored to its effective position and thereafter it is impossible to exceed full-load delivery of fuel until after the engine has been stopped. *Patent No.* 701662. *Robert Bosch G.m.b.H.* (*Germany*).

Air-cooled cylinders

IT is commonly necessary to interrupt the transverse fins of an air-cooled cylinder to provide clearance for holding-down bolts, push rods and inlet or exhaust pipes. As a consequence the cylinder lacks a uniform rigidity and, when heated during operation of the engine, expands unevenly. The fewer the interruptions and the more irregularly they are distri-

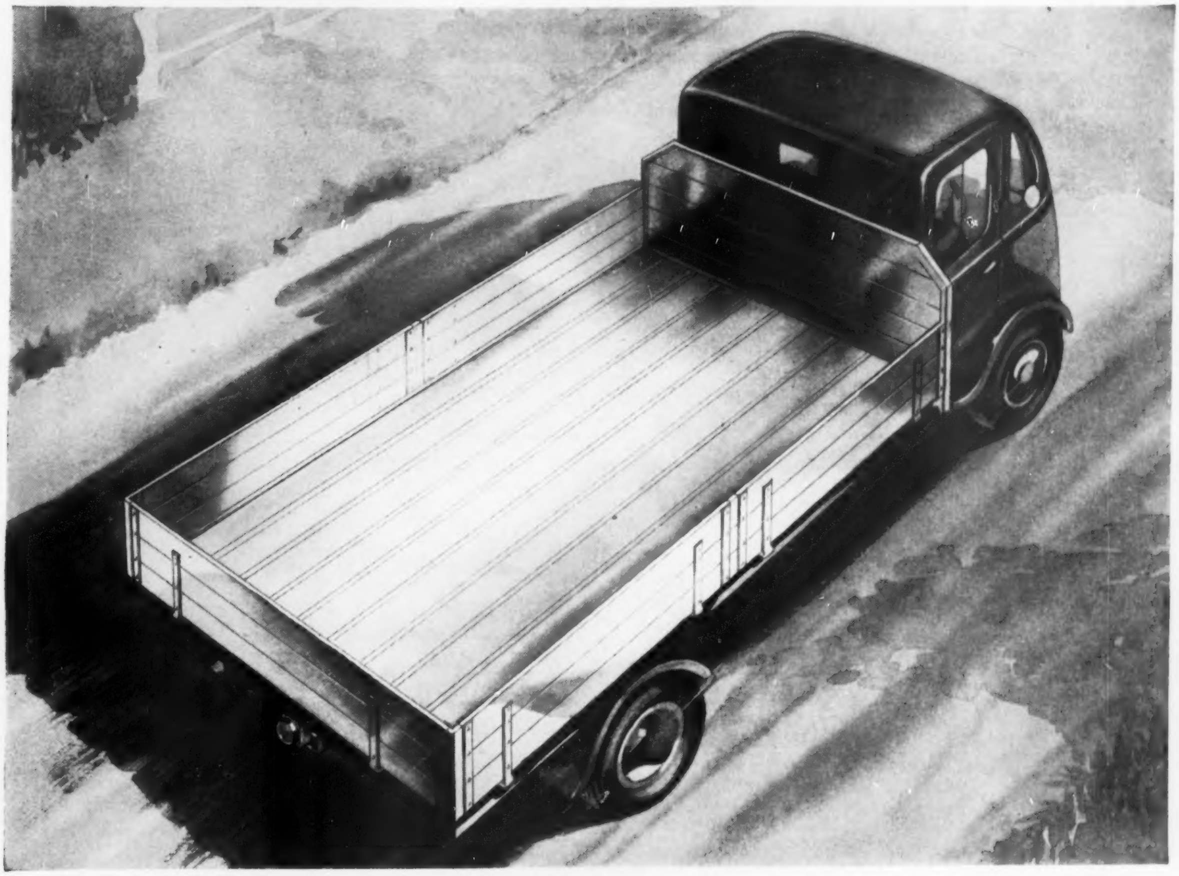


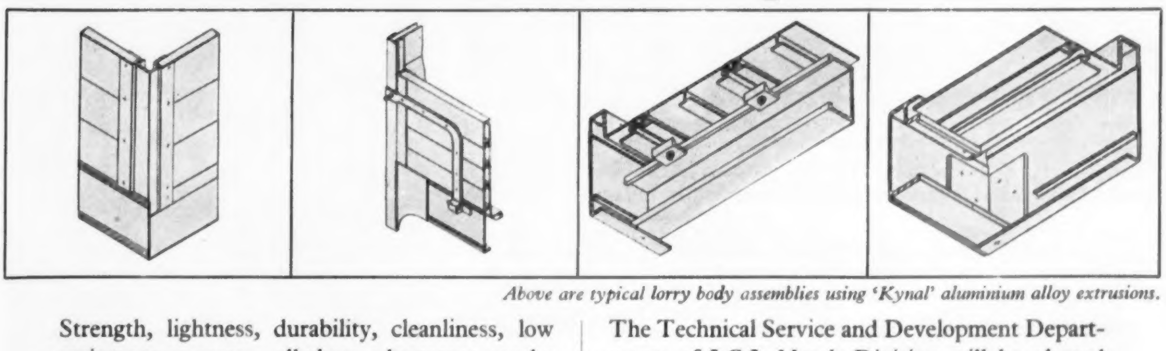
No. 701163

buted around the cylinder the greater is the subsequent wear of piston and bore.

According to the invention these disadvantages are obviated by dividing the fins between interruptions so that the radial intervals are approximately uniform and the cylinder, on expansion, will deform to a regular polygon. The slots in the fins should, of course, be of a similar depth to that of the interruptions. For example, the fins of cylinder A are gapped at four points to clear holding bolts B and the intervening portions of fin are slotted at C to provide a regular pattern of interruptions.

The port branch of the cylinder of a piston-controlled two-stroke engine constitutes a substantial transverse reinforcement likely to cause uneven expansion. This is particularly unfavourable when the port is bridged since the bar or bars D may become bowed towards the interior of the cylinder when heated. As a remedy it is proposed to bifurcate the port branch and slot the fins above, and also possibly below, the port. In this instance the depth of the slots should decrease progressively from the port, as shown at E. For a simple port the rigidity may be reduced by slotting it, longitudinally of the cylinder, to the appropriate depth. The slot is then covered by the inserted exhaust pipe. *Patent No.* 701163. *Klockner-Humboldt-Deutz A.G.* (*Germany*).





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b One change of speed forward and one reverse or two speeds in either direction.
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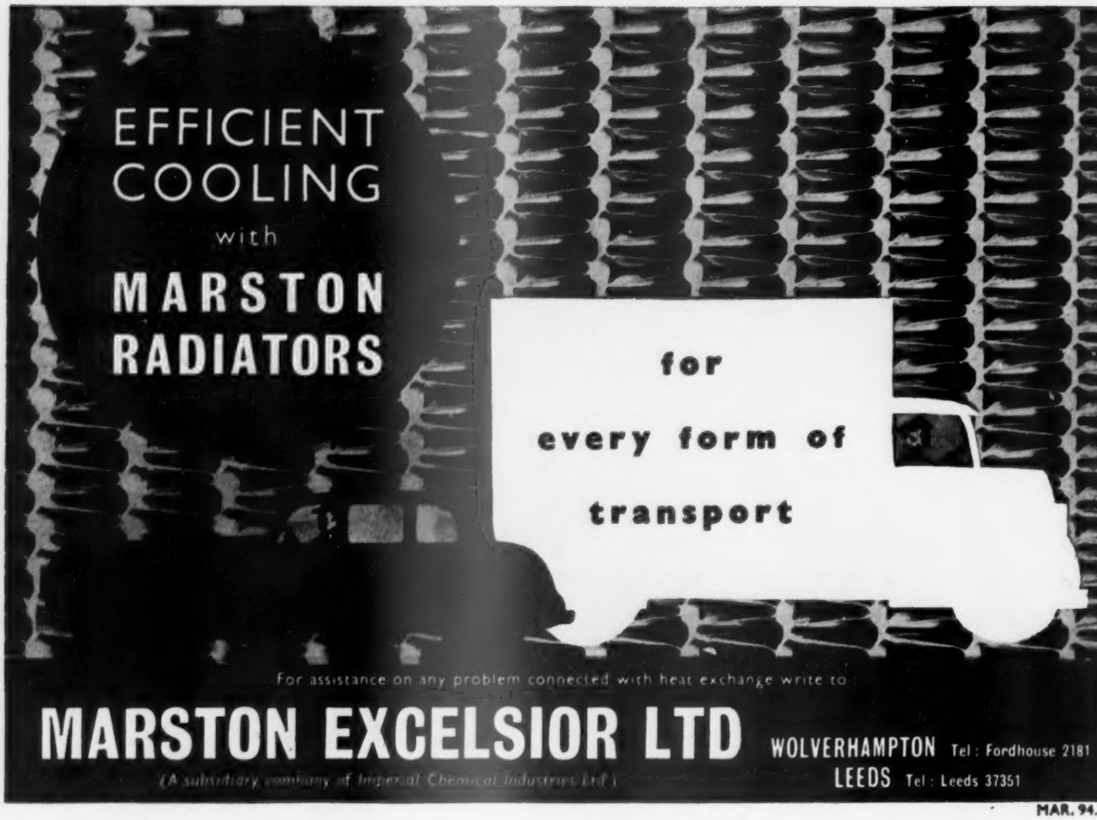
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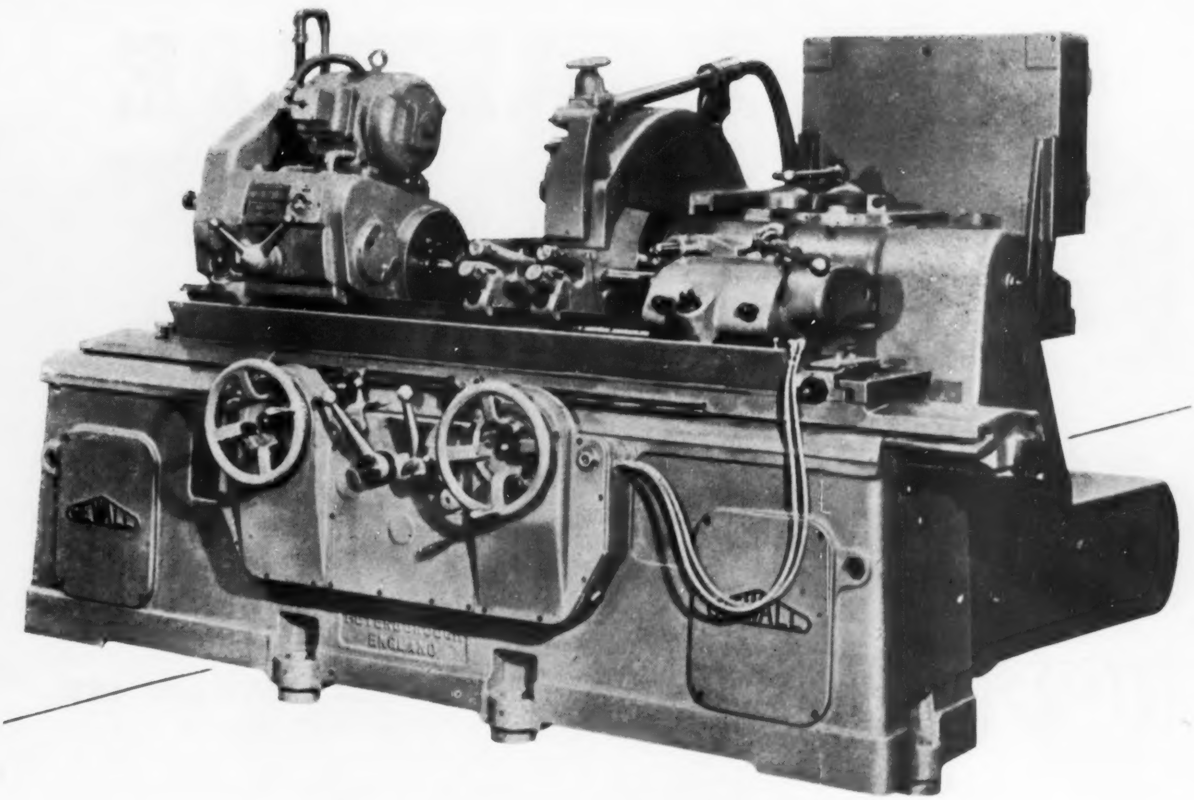

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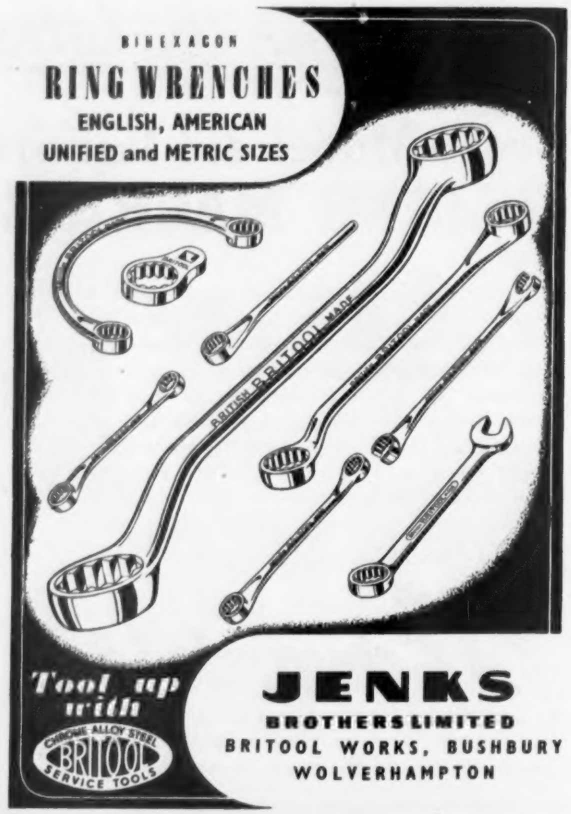

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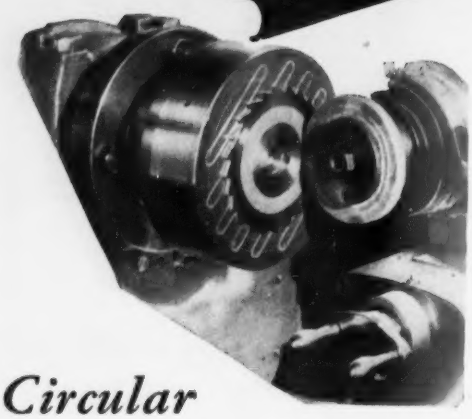

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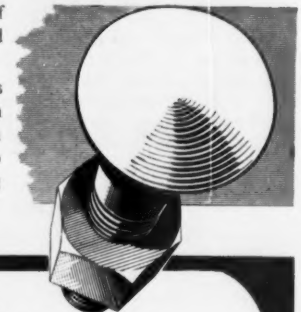

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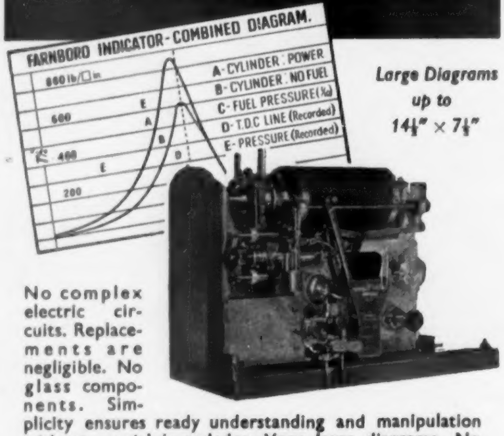

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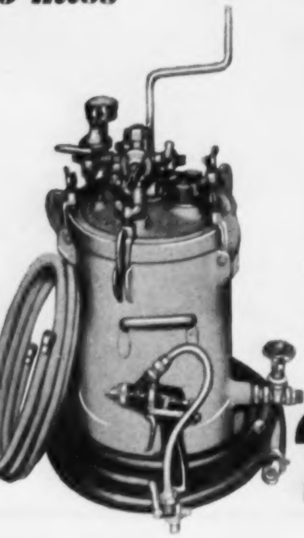

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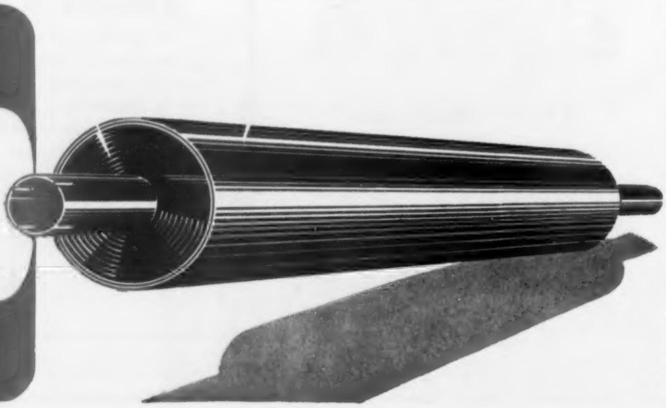

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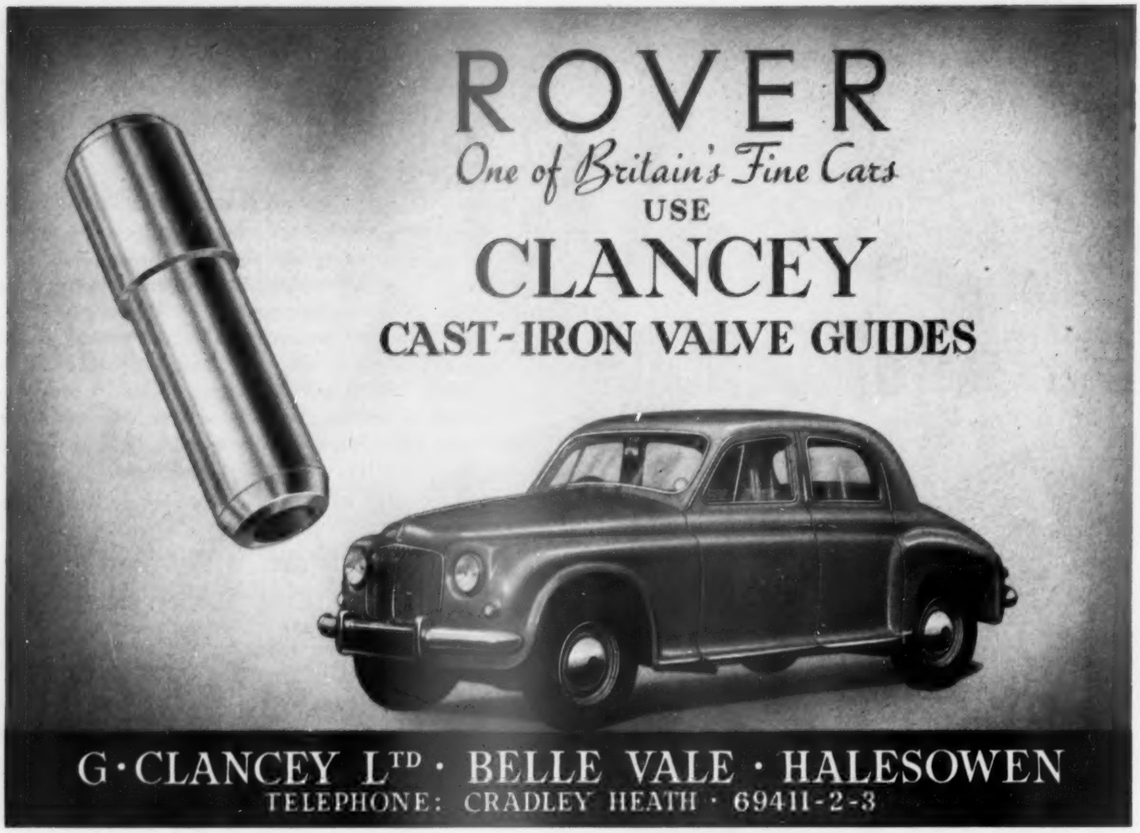
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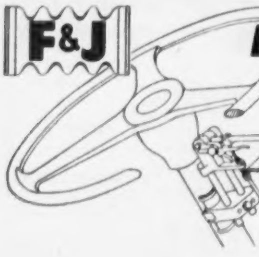

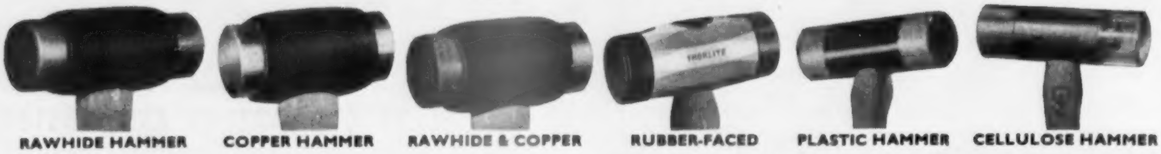
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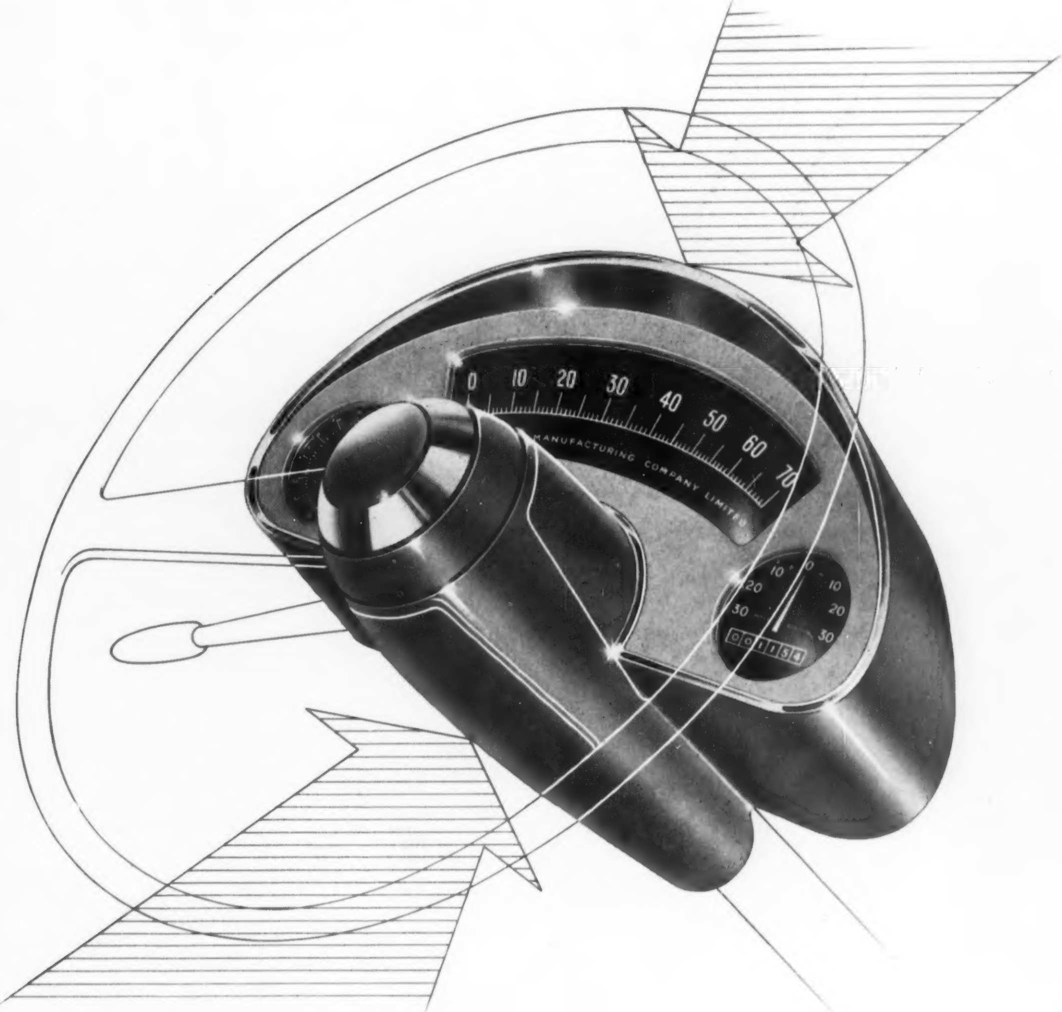

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